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CONTENTS

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CONDENSING PLANT

CONDENSING PLANT

A COMPLETE TREATISE ON THE PRINCIPLES
AND DETAILS OF CONSTRUCTION OF
MODERN STEAM CONDENSING APPARATUS

FOR
DESIGNERS, USERS AND STUDENTS

BY
R. J. KAULA, M.I.E.E.

AND
I. V. ROBINSON, WH.Sc.
ASSOC M INST.C.E., M.J.MECH E., M AMER.S.M.E.



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PREFACE

THE name of James Watt is associated in the popular mind with the invention of the steam engine. It is, of course, well known that whereas Watt's genius was responsible for almost innumerable improvements in its design, he was not the actual inventor of the steam engine. On the other hand, he was, in fact, the inventor of the steam condenser as a unit, in contrast to previous practice whereby condensation of the steam was effected within the engine cylinder. His first patent describing this invention is No. 913, of 1769; and in the second claim he gives the name of Condenser to the separate vessel in which the steam is condensed. The third claim is notable in that it records the use of an air pump to remove "whatever air or other elastic vapour is not condensed by the cold of the condenser."

From Watt's days onward we can trace a steady progress in the design of condensers and their auxiliaries, but the rapid development of the steam turbine industry at the beginning of the present century gave a new impetus to the study of condenser problems. The then known forms of reciprocating engines could not effectively utilize vacua above 26 ins. to 27 ins., but The Hon. Sir Charles Parsons, F.R.S., and later turbine builders realized the advantage of extending the steam range in a turbine to the highest vacua attainable.

As a result, the whole subject of the value of efficient condensing apparatus took on a new and enlarged aspect. From being an inconsiderable auxiliary to a steam engine, the modern turbine condensing plant has developed into a vital link in a steam power plant, ranking in importance with the boilers and the turbine itself, and frequently occupying a greater volume than does the turbine that it serves. The progress achieved is largely due to the experimental work and writings of D. B. Morison, R. L. Weighton, G. A. Orrok, Professor Josse, Maurice Leblanc, and others.

The various items that constitute condensing plants are subjected to comparatively light mechanical stresses whose value can be calculated by simple mathematical methods. It is, therefore, quite natural that serious mechanical breakdowns and failures should be practically unknown. The failure of an auxiliary due to minor causes, such as hot bearings or defective glands, may be a source of serious inconvenience, but is at any rate not of a destructive nature, such as the bursting of a boiler or of a turbine rotor.

A complete and sudden failure of the vacuum has occasionally been the cause of a serious turbine breakdown, and every attention must be paid to the reliability of the auxiliaries and their drives and to the problem of the effective straining of the circulating water. Since the capital value of the condenser auxiliaries represents a small percentage of that of the complete unit that they serve, there is a *prima facie* case for safeguarding the plant as a whole by installing the auxiliary pumps in duplicate. On the other hand, the difficulty of providing the necessary pipelines and valves for such an arrangement is always considerable and generally leads to a cramped condenser lay-out which is a most undesirable feature in any power station. These objections do not apply to the same extent where the plant is so arranged that the condenser is divided into two sections each provided with its own auxiliaries and where the combined capacity of the two sections represents the full capacity of the plant. In such a case, one section of the plant can be shut down at any time without interfering with the effective operation, at reduced vacuum, of the remaining section.

From the time of the commencement of the steam turbine industry there has been a continuously progressive movement, raising the vacuum in condensers for the purpose of utilizing the energy available in the lower steam range. The increasing difficulty and expense of building high-speed turbines effectively to use high vacua, combined with the absolute limits imposed by cooling water conditions, have caused turbine builders and users, latterly to look in the opposite direction, namely, the raising of the initial steam pressure and temperature for a more convenient means of raising the thermal efficiency of steam-power plant. We can therefore look to a period when condenser designers will concentrate their efforts on the more effective utilization of materials and on the simplification and on the improvement in the mechanical efficiency of the auxiliaries. In this respect the modern air pump gives most scope for improvement since the mechanical efficiencies so far achieved are still very low.

In the development of both hydraulic and steam ejector types of air pump reliability and space efficiency, coupled with capacity as expressed in terms of maximum vacuum attainable for a given air quantity, have outweighed the question of mechanical efficiency and of power consumption. The earlier hydraulic air pumps owed their success mainly to the fact that they lent themselves readily to direct driving by electric motors or by small steam turbines; in other words, to the fact that they were rotary machines running at comparatively high speeds. Their power consumption was often

a good deal higher than that of a reciprocating mechanical pump designed for the same duty, and even to-day it is doubtful if hydraulic and steam jet pumps show any marked advance in mechanical efficiency over reciprocating pumps as built twenty years ago. The present success of the steam ejector air pumps is, in turn, probably due to the absence of a mechanical driving agent, and to the convenience of handling small diameter steam pipes instead of water pipes that become very cumbersome in hydraulic air pump installations of large capacity.

If we examine the later designs of surface condensers for large duties it will be apparent that in so far as they show a departure from the earlier and simplest forms, the underlying object of the improvement is throughout in the direction of shortening the steam path. These modifications show substantial gains as expressed in the form of a higher overall heat transmission coefficient K , which is simply an expression of the duty performed for a given amount of cooling surface and for equal water conditions and air pump capacity.

The advantage obtained appears to be due to the favourable concurrent working of several factors. Thus a short steam path design will present a relatively large surface that is free from dropping condensate. The reduced drop in pressure across the condenser means that the absolute maximum vacuum to be obtained in the condenser is reduced for a given vacuum at the condenser inlet. The fact that this increases the mean temperature difference between steam and cooling water is a matter of minor importance, as the gain is comparatively small but the resultant relative increase in air pump capacity and consequent relief to the condenser in the matter of air cooling is probably the most substantial factor in producing the improvement obtained in designs of this description.

In the present volume the authors have endeavoured to treat the subject from the standpoint of the user as well as that of the designer and student. They have dealt in greater detail with condensing plant problems appertaining to land practice than with those relating to marine practice, as they have been able to write with greater personal experience in that direction.

The authors' thanks are due to the numerous Condenser makers in this country and abroad, to whom reference is made in these pages and who have very kindly supplied them with descriptive matter relating to their manufactures as well as to their many friends engaged in the industry, whose advice in many directions has been of the utmost value.

CONTENTS

	PAGE
PREFACE	V
CHAPTER I	
INTRODUCTION	1
Surface condensers—Jet condensers—Ejector condensers—Atmospheric condensers—Performance of condensers	
CHAPTER II	
THE ECONOMY OF VACUUM CONDENSERS	5
Saving in steam consumption—Financial saving—Capital Expenditure—Optimum value of vacuum	
CHAPTER III	
CALCULATION OF QUANTITY OF COOLING WATER REQUIRED	11
Heat contents of exhaust steam—Cooling water required	
CHAPTER IV	
SURFACE CONDENSERS—GENERAL CALCULATION OF COOLING SURFACES	15
Principles of design—Calculation of surface area—Approximation	
CHAPTER V	
SURFACE CONDENSERS — DETERMINATION OF THE HEAT TRANSMISSION COEFFICIENT K	20
Subdivision of R and K —Value of R_s —Effect of incrustation on heat transmission—Determination of R_1 —Effect of presence of air in condensers—Numerical value of R_1 —Determination of R_s —Critical water velocity—Suitable velocity for flow of water—Viscous annulus in turbulent flow—H. M. Martin's calculation of R_s —Considering the flow as entirely viscous—Taking the flow as turbulent	
CHAPTER VI	
SURFACE CONDENSERS—COOLING SURFACE FOR AIR AND CONDENSATE	33
Effect of presence of air on temperature of condensation—The necessary drop in temperature—Calculation of steam quantity carried over by the air pump—Heat transmission coefficient for air—Mean temperature difference—Surface allowed for air cooling—Surface area for cooling the condensate—Division of total surface	

CHAPTER VII

SURFACE CONDENSERS—CALCULATION OF DIMENSIONS . . .	PAGE 43
Number of water passes—Ratio of diameter to length—Effect of modifications	

CHAPTER VIII

SURFACE CONDENSERS—CALCULATION OF WATER HEAD . . .	48
Loss of head due to friction—Calculation of frictional resistance in tubes—Examples showing nature of error due to use of Simplified formulae—Friction loss in boxes and tube ends—Circulating water-pump characteristics	

CHAPTER IX

SURFACE CONDENSERS—CONSTRUCTIONAL DETAILS . . .	54
Condenser tubes—Manufacture of tubes—Testing tubes during manufacture—Calculating bursting pressure of tubes—Determining deflection of tubes—Expansion of tubes in condenser—Vibration of tubes in condenser—Tube plates—Thickness of tube plate required—Typical forms of construction—Tube glands—Shells—Stresses acting on shell—Required thickness of shell—Section of exhaust flange—Water heads and doors—Total load on doors	

CHAPTER X

SURFACE CONDENSERS—TYPICAL DESIGNS	75
Contraflo condensers—Principles of design—Contraflo marine type condensers—Contraflo atmospheric condenser—The Weir uniflux condenser—Weir auxiliary condensers—Worthington-Simpson condensers—Worthington Pump and Machinery Corporation—The Wheeler Condenser and Engineering Co.—Under-type Wheeler condenser—Wheeler waterworks condensers—Alberger Spiroflo condensers—The Ingersoll Rand condensers—External cooler—Ingersoll Rand tube specification—Escher Wyss condensers—The Gervenbroich condensers—Brown-Boveri O.V. condensers—Metropolitan-Vickers condenser—The "central flow" condenser—"Central flow" condenser with divided water paths—Mirrlees-Watson condensers—The Société des Condensateurs Delas of Paris and Saint Ouen	

CHAPTER XI

CORROSION AND EROSION—SURFACE CONDENSER—FOULING OF CONDENSER TUBES	110
Causes of tube corrosion—Electrolytic corrosion—Protection against electrolytic corrosion—The "Cumberland" protective system—The coating of tubes—Causes of tube erosion—The fouling of tubes—Formation of scale in the tubes—Rate of formation of scale—The scale-forming process—Effect of scale on efficiency of condenser—Cleaning without interrupting service—Brown-Boveri continuous service condenser—Advantages of continuous service—The Hulsmeyer self-cleaning condenser—Mirrlees-Watson vertical condensers—The Mirrlees-Watson tube cleaner	

CONTENTS

xi

CHAPTER XII

	PAGE
EVAPORATIVE CONDENSERS	127

The Ledward and Beckett condenser—Relative advantages of both types of condensers—Economy in auxiliary power due to evaporative condenser—The use of evaporative condensers—The "Ramsey" condenser

CHAPTER XIII

JET CONDENSERS	138
--------------------------	-----

General considerations and types—Loss of condensate—Power consumption—Shape—Types of jet condensers—The low-level type—The barometric type—Counter current and parallel flow jet condensers—Methods of overcoming disadvantage of parallel flow type

CHAPTER XIV

JET CONDENSERS—CALCULATION OF WATER HEAD	144
--	-----

Water head on head pump—Water head on tail pump

CHAPTER XV

JET CONDENSERS—TYPICAL DESIGNS	147
--	-----

Metropolitan-Vickers jet condenser—The M.A.N. jet condensers—The Alberger jet condensers—The Alberger Spiroflo condenser—The Alberger barometric condensers—The Mirrlees-Watson jet condenser—Worthington-Simpson jet condenser—Worthington-Simpson concentric design—The Hick-Breguet low level jet condensers

CHAPTER XVI

JET CONDENSERS—EJECTOR TYPE	162
---------------------------------------	-----

• The Koerting ejector condenser—Twin Koerting multi-jet condensers—Performance of Koerting plant—Mirrlees-Watson single jet condenser—The Rees-Roturbo rotary jet-condenser

CHAPTER XVII

AIR PUMPS—RECIPROCATING TYPE CALCULATIONS	173
---	-----

Pump capacity—Effect of clearance space—Variation of air pump efficiency

CHAPTER XVIII

AIR PUMPS—STEAM EJECTOR TYPE CALCULATIONS	182
---	-----

Supersaturated steam—Dryness fraction of the steam—Action in entraining chamber—Conditions at entry to diffuser—Motion in diffuser—Mixing in diffuser—Conversion of kinetic to pressure

energy—Adiabatic compression of a mixture—Determining area of diffuser—Use of Callendar steam diagram—Ejector with water jacket diffuser—Conditions on establishing a vacuum—Calculating the proportions of a steam jet—Comparison between isothermal and adiabatic compressor—Efficiency of each system—Importance of efficiency of steam jet—Action of the ejector

CHAPTER XIX

AIR PUMPS—HYDRAULIC TYPE CALCULATIONS . . . 211

Hydraulic air pumps—Leblanc rotary pump

CHAPTER XX

AIR PUMPS—DESCRIPTION OF RECIPROCATING TYPES . . . 220

The Edwards air pump—The Parsons compound air pump—Parsons vacuum augmentor—Wet and dry air pumps—Dry air pumps—The Worthington feather valve—Pump valve of the Mirreles Watson Co—The Weir "Dual" air pump

CHAPTER XXI

AIR PUMPS—DESCRIPTION OF STEAM EJECTOR TYPES . . . 237

Steam ejector air pumps—Development—The Breguet "Ejector"—The Mirreles-Leblanc ejector—Experimental designs to facilitate starting—Metropolitan Vickers Leblanc type ejector—Mirreles Watson "Central Condenser" type—The Wheeler multiple jet ejector—Heat economy in second stage—Combined after-condenser—Vacuum trap—The Wheeler "Radojet" ejector—The Kinetic air pump—The Delas ejector—The Ljungstrom ejector—The Worthington ejector—The Ingersoll-Rand ejector—The Weir ejector—Brown-Boveri ejector—Pipework arrangement—Meldrum ejectors

CHAPTER XXII

HYDRAULIC AIR PUMPS . . . 271

Leblanc pump—The Scanes pressure tank—The Worthington hydraulic pump—Escher Wyss hydraulic extractor—Müller air extractor—Means of preventing flooding—Müller ejectors with Hulsmeier system condensers—Rejection of air and loss of condensate—The Kinetic rotary pump—The Brown-Boveri hydraulic ejector—Two stage extraction pump of the Contraflo Co—The Wheeler rotary air pump

CHAPTER XXIII

AIR PUMPS—COMPARISONS OF ROTARY AND EJECTOR PUMPS . 300

Methods of heating the condensate—Advantages of each type of pump

CONTENTS

xiii

CHAPTER XXIV

	PAGE
AIR PUMP DRIVES	304
Motor or turbine drive for auxiliary pumps—Considerations of capital cost—Considerations of starting conditions—Heating condensate by steam tapped from the main turbine—Heating condensate by exhaust steam—Heat balance	

CHAPTER XXV

TESTS ON CONDENSING PLANT AND AUXILIARIES	310
Müller hydraulic air extractor—Hydraulic ejector more efficient than steam ejector—Time to establish vacuum—Tests on steam ejectors—Effect of varying steam pressure—Conditions on starting up an ejector—Axial position of jets—Variation of steam ejector capacity with vacuum—Steam consumption of an ejector—Test on a Delas ejector—Tests of a Hicks-Breguet ejector—Plant by Worthington-Simpson, Ltd—Plant by Richardsons, Westgarth & Co, Ltd—Temperature corrections of mercury column readings—Variation of pressure at steam inlet to condenser—Method of equalizing pressure at steam inlet to condenser—Air quantities—Recommended air capacity for various plants—Manchester tests—Value of K—Plant by C. A. Parsons & Co, Ltd—Plant by Alberger Condenser Co—Plant by Wheeler Condenser & Engineering Co, Ltd—Plant by the Murrlees-Watson Co, Ltd—Plant by Schutte & Koerting—Method of recording routine readings	

CHAPTER XXVI

CLOSED FEED SYSTEMS AND DE-AERATING PLANTS	354
Effect of air in steam—Air contents of aerated water—The surface condenser as de-aerator—Importance of closed feed system—Advantage of closed feed system from boiler standpoint—Air contents of condensate—Details of de-aerating plant—The Contraflo closed feed system—Contraflo de-aerating systems—The Murrlees Watson closed feed system—Ellhott de aerating system—Weir closed feed system—Brown Boveri feed systems—Closed Weir type condensate meter—Method of ascertaining oxygen contents	

STEAM TABLES	392
INDEX	395

INSET PLATES

FIG		BETWEEN PAGE
76	General Arrangement of Hick-Breguet Low-level Jet Condensing Plant	160-161
214	Metropolitan-Vickers Marine Turbine and Condenser	344-345
217	General Arrangement of 26,000 sq ft Plant	350-351
223	The Contraflo Closed Feed System	366-367
TABLE I	Quantity of Cooling Water Required per 1,000 lbs of Steam, in Gallons per Minute	394-395

CONDENSING PLANT

CHAPTER I

INTRODUCTION

A STEAM condenser may serve one or both of two purposes. Firstly, it may be designed solely for the recovery of water that has been converted into steam at a pressure above that of the atmosphere, and whose pressure has been allowed to fall to that of the atmosphere in an engine or in a steam turbine. Secondly, it may be intended to produce a vacuum so as to extend the range of expansion of the steam below the atmospheric pressure. Lastly, it may be designed to embody both these functions. In the first and last types of condenser the steam needs to be separated from the cooling medium—usually water and such condensers are known as atmospheric and vacuum surface condensers. The water formed as the result of condensation is termed the condensate, and the solid medium placed between steam and cooling water which generally takes the shape of tubes is called the cooling surface.

Steam entering a condenser carries with it a proportion of air and other gases from which feed water is never entirely free. In all vacuum condensers these gases have to be removed by means of an air pump, otherwise they would accumulate in the condenser and destroy the vacuum.

The combination of condenser and air pump together with the cooling water and condensate pumps, the pipes, valves and accessories required for the working of the combination constitute a condensing plant. The term "condenser" which, in fact, only covers the body in which condensation takes place, is frequently used to indicate the whole condensing plant of which it forms the most important part.

Surface Condensers generally consist of a cast iron or mild steel shell, fitted with tube plates at either end which carry a number of tubes forming the surface. In most cases steam is admitted to the condenser through an opening in the top of the shell, and water is passed through the tubes. In some cases, however, more especially where the condenser is used in conjunction with a pumping engine, the reverse arrangement is adopted, that

is, the water passes over the outside of the tubes and the steam is allowed to pass through the inside of the tubes. In yet another form of surface condenser, known as evaporative condensers, the shell is omitted altogether, and the steam is led to the condenser inside the tubes. Water is sprinkled over the outer surface, and cooling is effected mainly by evaporation of this water in the surrounding atmosphere.

Jet Condensers. In the second or jet type of condenser the steam is brought into immediate contact with the cooling water, which has to be divided into a large number of water jets to expose a maximum water surface to the steam and to ensure the closest possible intermingling between steam and water.

Ejector Condensers. In certain types of jet condensers the water jets are so arranged as to carry away the air. They combine the action of the ordinary jet condenser with that of an hydraulic air pump, and are known as Ejector condensers.

Atmospheric Condensers. The use of atmospheric condensers is limited to conditions where the recovery of the boiler feed water is a matter of importance, but where the maintenance of a vacuum in the engine or the turbine exhaust is not required, that is to say, where the capital cost involved in installing a vacuum plant, together with the additional power consumption of the auxiliary pumps, will not show an adequate return in the shape of saving in steam and coal consumption. They find their largest field of application in conjunction with winch and other auxiliary engines on board ship, where all boiler feed make-up water has to be obtained by evaporation, and where a vacuum cannot be economically used or maintained. Incidentally, they serve a second purpose in that the condensate of a well-designed condenser of this type should be at a temperature within a few degrees of atmospheric boiling point.

The comparative advantages of jet and surface vacuum condensers will be discussed in a later chapter, but it may here be said, that where a suitable supply of boiler feed water is not available, a surface plant must almost invariably have preference over a jet plant, notwithstanding its higher first cost.

Performance of Condensers. Since the primary purpose of a vacuum condensing plant is to maintain a given absolute back-pressure at the engine or turbine exhaust flange, the condenser performance should be measured by the pressure in lbs. per square inch absolute maintained at the exhaust flange. The usual form of vacuum gauge, however, measures the pressure in the condenser in relation to the surrounding atmospheric pressure. The standard atmospheric pressure is taken at 14.730 lbs. per sq. in. or 30 ins.

of mercury at 32° F., and vacuum gauges are usually calibrated to this pressure in inches of mercury. Where the metric system is used they are marked to indicate the vacuum as a percentage of the normal atmospheric pressure. It has therefore become customary to refer to condenser performance in inches of mercury, or as a percentage, and in doing so it is understood that the measurement refers to normal barometric pressure. To avoid misunderstanding it is usual to quote the barometric pressure in parenthesis, thus, "vacuum = 26 ins. (Bar. 30 ins.)." In the present volume this precaution is not called for, and has been omitted.

Since the barometric pressure varies with the altitude and with every atmospheric change, and gauges simply register the difference in pressure inside and outside the condenser, gauge readings without a reading of the barometer at the time they are taken are useless. Unfortunately, this fact is not always appreciated, and it is still quite common to find vacuum gauge readings quoted without reference to the barometer at all.

To enable us to arrive at the right manner of correcting a vacuum reading for variations in barometric pressure, we shall consider the action of the simplest form of vacuum gauge, namely, an open-ended U tube, having legs 30 ins. high and containing 30 ins. of mercury, one leg being connected to the condenser and the other being open to the atmosphere. If the condenser is out of action then the mercury will, of course, be evenly divided in the two legs and stand 15 ins. above the bottom. If the barometric pressure is 30 ins. and a complete vacuum is maintained in the condenser, then the mercury will rise to the top of the leg connected to the condenser and will fall to the bottom of the atmospheric leg. In other words, the atmospheric pressure on one side will be balanced by the mercury column in the other leg. The calibration between 0 in. and 30 ins. of vacuum extends therefore over the upper 15 ins. of tube, and the scale is represented by $\frac{1}{2}$ in. to 1 in. vacuum. Assume that the atmospheric pressure drops to, say, 28 ins., and that the scale indicates a vacuum of 20 ins., that is to say, the mercury column on the condenser side is 15 ins. + 10 ins. = 25 ins. above the bottom, and let the actual back-pressure prevailing in the condenser be X. Then the balance between the two legs is made up of 28 ins. of atmospheric pressure, plus 5 ins. of mercury on one side and of 25 ins. of mercury plus X on the other side, so that $X = 33 \text{ ins.} - 25 \text{ ins.} = 8 \text{ ins.}$ against the indicated reading of 10 ins. The correction is therefore from 20 ins. to 22 ins., and it can be similarly shown in every case that the correction must be made by adding to or deducting from the vacuum reading the difference of the barometric

reading below or above the normal pressure of 30 ins., at which the gauge has been calibrated. If, on the other hand, the vacuum is read on a mercury column registering inches of back-pressure or lbs. per square inch absolute, no such correction is required.

The variations in pressure of the outer atmosphere do not concern the actual performance of the plant, except in their effect on the work to be performed by the air pumps, in that the greater the atmospheric pressure the greater the work of the air pump in compressing the air to be removed from the condenser. Naturally, the saving in steam consumption due to the use of vacuum condensers is dependent on the barometric pressure prevailing; for if in a given locality the normal barometric pressure is only, say, 23 ins., then an engine exhausting to atmosphere is already working under identical conditions to those prevailing where an engine exhausts into a condenser registering 7 ins. of vacuum when the atmospheric pressure is 30 ins.

Some further questions relating to vacuum gauge corrections are discussed in Chapter XXV.

CHAPTER II

THE ECONOMY OF VACUUM CONDENSERS

A POUND of steam at a pressure of 150 lb. (gauge) per sq. in. superheated to a total temperature of 550° F., contains 1303 B.Th.U.'s. If this steam is expanded adiabatically to atmospheric pressure then 204.5 B.Th.U.'s are made available for useful work. If the

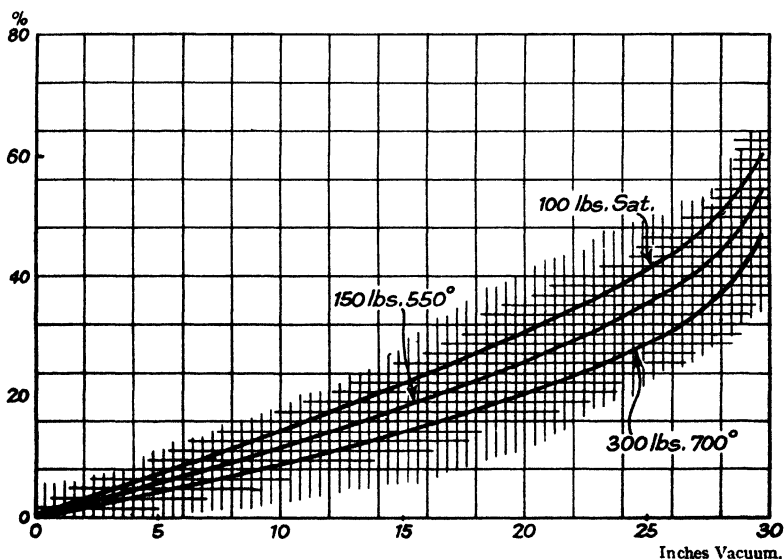


FIG. 1.
THEORETICAL SAVING DUE TO VARYING VACUA FOR
DIFFERENT INITIAL STEAM CONDITIONS.

same steam is expanded to a vacuum of 29 ins., 400.23 B.Th.U.'s will be so available. That is to say, given the above initial steam conditions in an ideal engine, the steam consumption will be 49 per cent lower when working condensing with a 29 ins. vacuum than when working non-condensing.

Saving in Steam Consumption. Fig. 1 shows the theoretical percentage saving in steam consumption due to varying vacua from 0 to 29.1 ins. for three widely differing initial steam conditions. In reciprocating engine practice, the percentage gain is as a rule below the theoretically possible, since the thermodynamic efficiency

is higher above atmospheric pressure than below. In turbine practice this does not hold especially as regards the lower range of vacua, so that the theoretical saving does not necessarily represent a maximum. Where for instance, a well-designed low-pressure turbine is used in conjunction with a comparatively inefficient engine such as a winding or a rolling mill engine, it is by no means unusual to find the saving in steam consumption due to the vacuum exceed the theoretical gain.

Financial Saving. The gain in steam consumption represents only one of a number of factors which require to be considered in determining the economy resulting from the maintenance of a given vacuum. We are concerned to show a return in money for any additional capital expenditure involved. This return is represented mainly by a saving in expenditure on fuel and on labour in handling the fuel.

To determine this saving we have on the credit side the saving due to the reduction in steam consumption, and on the debit side the loss due to power consumption of the condenser auxiliaries and the loss in fuel entailed due to the lower condensate temperature.

Capital Expenditure. In arriving at the capital expenditure due to the condensing plant we must take account of the additional cost of the prime mover, the cost of the condensing plant with all its pipework, and of the cooling towers and the land they occupy if such are required. When a natural supply of cooling water is available, there is generally a capital expenditure, and occasionally a very considerable one, due to pipe-lines, ducts and strainers. On the credit side we have a saving in the cost of the boiler plant and of its buildings and accessories such as coal and ash-handling plant, feed pumps, steam piping, and the like.

Optimum Value for Vacuum. Whilst it is a comparatively simple matter to arrive at a reasonably safe conclusion as to the economy or otherwise of working a given steam plant to atmosphere or condensing, it must always be a difficult question to decide on the most economical vacuum for any given set of conditions. A number of carefully considered contributions to this subject have been published from time to time,* but even where an individual case is fully investigated one can only arrive at a fair approximation owing to the fluctuations in load factors, price of coal, etc., which cannot be exactly predetermined. Let us take as a concrete example the case of a 3,000 kw. turbo-alternator running at 3,000 r.p.m. and supplied with steam at a pressure of 150 lbs. gauge at

* See Prof. G. Gerald Stoney, F.R.S., Min. of Proc. of the Inst. of Mech. Engs., 20th and 24th Nov., 1914, "The Effect of Vacuum in Steam Turbines."

a temperature of 550° F. exhausting to atmosphere or into a surface condenser. We shall further assume that cooling towers are required to reduce the temperature of the circulating water to 75° F. We can then work out a series of balances for conditions of non-condensing and condensing with vacua of 27½ ins., 28 ins. and 28½ ins. respectively based upon a 50 per cent load factor.

TABLE I
FINANCIAL BALANCE FOR VARIOUS VACUA FOR A 3,000 KW.,
3,000 R.P.M. TURBO-ALTERNATOR UNIT.

VACUUM AT FULL LOAD	0"	27-5"	28-0"	28-5"
<i>Capital Cost—</i>				
Boilers, etc.	£21,550	£12,200	£11,900	£11,600
Turbo-alternator	£12,000	£13,000	£13,000	£13,000
Condensing plant	—	£3,200	£3,600	£4,300
Cooling towers	—	£1,720	£2,150	£3,700
Total capital cost	£33,550	£30,120	£30,650	£32,600
Saving over 0" vacuum	—	£3,430	£2,900	£950
Annual value of above. (Depreciation, interest, and maintenance at 18% per annum)	—	£618	£522	£171
<i>Coal Costs—</i>				
Steam consumption on mean load (½) lbs. per the hour	26-0	13-1	12-7	12-35
Tons of steam per annum 50% load factor	152,400	76,700	74,400	72,400
Tons of coal per annum (11,000 B.Th.-U.'s per lb, 70% boiler efficiency feed at 180° F)	22,600	11,360	11,000	10,710
Condensate temperature (5° below vacuum), degrees F	180°	103°	96°	87°
Extra coal to raise feed to 180° (tons per annum)	—	770	812	875
Net saving in coal against 0" vacuum corrected for feed heating loss, in tons per annum	—	10,470	10,788	11,015
Power consumption of condenser auxiliary pumps and feed pump in B.H.P.	40	100	123	221
Extra power over 0" vacuum, B.H.P.	—	60	83	181
Corresponding coal consumption, tons per annum	—	203	261	580
Net coal saving over 0" vacuum, tons per annum	—	10,267	10,527	10,435
Value of coal saved at £1 per ton	—	£10,267	£10,527	£10,435
Total annual saving over 0" vacuum	—	£10,875	£11,049	£10,606

The non-condensing figures have been given as a basis for comparison only. Conditions under which a non-condensing turbine

of this size could be employed could only arise if the exhaust steam is utilized for heating purposes.

The capital charge for the boiler plant should include that of the boilers and superheaters, economizers, coal-handling plant, feed pumps, etc. It may be argued that the loss or gain in steam consumption due to variations in vacua are not reflected in the purchasing of boiler plant, and consequently that their capital cost should not be taken into account in calculations of this kind. It is true enough that a buyer does not specify his boiler capacity on the bare basis figure of his engine or turbine steam consumption, but allows an ample margin. On the other hand, there is no reason why this margin should be larger if the vacuum is to be high or vice versa, and it seems therefore logical to include this factor in our calculations.

No alteration in the cost of the turbine is shown for the three conditions of vacuum, as for the size of machine under consideration one and the same standard frame would be used, the only difference due to the variations in vacua being in the nozzles and blade angles, especially of the last stage or stages. If turbines were redesigned throughout for each set of steam conditions that arises, a variation in cost would have to be allowed for, and at the same time the turbine would be able to take greater advantage of the additional energy available at higher vacua, i.e. there would be a greater saving in steam consumption than that shown. Such procedure, however, is only practicable in special cases where, for instance, exceptionally large units are to be built. In normal size units the case arises where the specified output approaches the maximum capacity of a standard frame. In such a case the designer would be obliged to select his next size of frame if a good economy is to be obtained on a high vacuum, whereas the smaller and less expensive frame would be large enough for a lower vacuum. Under such conditions we should again have a higher capital cost for the higher vacua plant, but the saving in steam and coal consumption would be correspondingly greater.

The rate of 18 per cent per annum to cover interest, maintenance, and depreciation would have been considered too high in pre-war times, but probably represents a fair average figure at the present time.

It is, as a general rule, impossible to foretell with any degree of accuracy either the load factor on which a given plant will run for the whole period of its useful life, or the average load.

On lighter loads there should always be an improvement in vacuum unless the air pump is underrated and the prime mover

should show a greater saving for any improvement in vacuum than at full load owing to its relatively larger low pressure blade area, etc.

The feed heating and economizer plant may be of such a nature as to make it unnecessary to take account of variations in condensate temperature. In our present example we have assumed no such provision, and it will be seen that the resultant loss in feed water temperature at higher vacua neutralizes approximately 10 per cent of the saving due to improved steam consumption. The price of coal is assumed to include the cost of stoking.

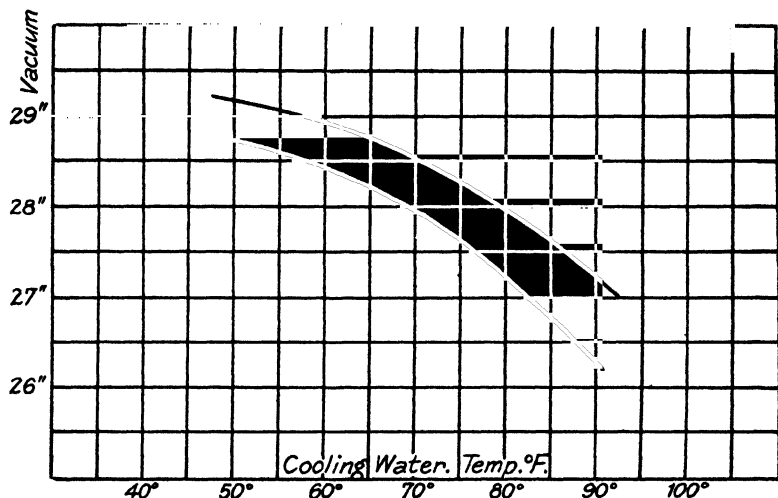


FIG. 2 (see p. 10).

NORMAL VACUUM FOR VARYING TEMPERATURES OF COOLING WATER

The condensing plant power consumption is based on the sum of air, circulating water, and condensate pump-power consumptions. The head on the circulating pump is taken as 45 ft., including delivery to the cooling tower inlet. The auxiliaries have been assumed to run on full load for 18 hours daily. It is rarely practicable to vary the auxiliary power consumption according to the load. With the object of attaining a minimum coal consumption it would be most desirable to reduce the water quantity passed at light loads: for instance, by means of a variable speed motor drive, but it is hardly possible to persuade operators to make proper use of such provisions, and it is better to assume constant power consumption at all loads. It will be seen that the auxiliary power consumption at 28½ ins. more than neutralizes the saving in steam consumption in comparison with 28 ins. vacuum, and we can therefore conclude that

so high a vacuum cannot be economically utilized where the mean cooling water temperature is 75° F., and where cooling towers are used regardless of any question of capital or coal costs.

In our present example it will be noted that there is a slightly higher net saving at 28 ins. than at $27\frac{1}{2}$ ins. vacuum. But the percentage difference is slight and would be reversed for a somewhat lower load factor.

In the foregoing example a number of minor items have been neglected which have no practical bearing on the resultant comparison such as cost of cooling tower make-up water, etc.

Whilst it is always desirable to work out a balance on the above lines for each case as it arises, the curves shown in Fig. 2 may be taken as limiting values for economical vacua for varying mean temperatures of cooling water. The upper limit applies to conditions of high load factor and coal cost and large units, and vice versa.

CHAPTER III

CALCULATION OF QUANTITY OF COOLING WATER REQUIRED

Heat Contents of Exhaust Steam. The heat (H_1) to be absorbed by the cooling medium is the total heat (H_e) contained in the steam less the heat (H_c) contained in the condensate, i.e. $H_1 = H_e - H_c$. If steam enters a condenser in its dry saturated condition, then the total heat H_e , according to Regnault's definition, is the sum of latent heat L , i.e. the heat required to convert the water into steam at the particular pressure (p), plus h , the heat required to raise the temperature of the water under pressure (p), from freezing point to the temperature corresponding to the boiling point of water at the pressure (p), i.e. $H_e = L + h$, then $H_1 = L + h - H_c$.

Since the temperature of the condensate is necessarily below the boiling point at p , it follows that H_1 is always greater than L by an amount corresponding to the difference between h and H_c . With a reasonably high condensate temperature, the value of $h - H_c$ is very small in proportion to L , and we shall not commit an appreciable error if we neglect the factor $h - H_c$, and put $H_1 = L$. The error is, in fact, generally more than neutralized by the assumption that the steam reaches the condenser in a dry saturated condition, whereas in fact it is usually wet or supersaturated.

The true heat contents of the steam reaching the condenser H_1 is equal to its initial heat contents H_2 , less the heat lost in the prime mover in radiation, bearing friction, etc., H_r , and less the heat equivalent to the useful work performed by the prime mover H_w .

$$H_e = H_2 - H_w - H_r$$

hence
$$H_1 = H_2 - H_w - H_r - H_c$$

TYPICAL EXAMPLES. To enable us to gauge the magnitude of the error introduced in taking $H_1 = L$, we shall work out two typical examples. Assuming a condenser to maintain a vacuum of 28 ins. when handling the steam exhausted from a turbine of 1,500 B.H.P. capacity and supplied (1) with dry saturated steam at a gauge pressure of 150 lbs. per sq. in., and (2) with steam at a gauge pressure of 250 lbs. per sq. in. superheated by 250° F. to a total temperature of 656·3° F.

In either case the latent heat is 1,035 B.Th.U.'s per lb.

In case (1) we shall assume an overall turbine efficiency of 65 per cent, and the turbine will consume 18,000 lbs. of steam per hour.

In case (2) the efficiency will be somewhat higher, say, 70 per cent, and the steam consumption will be 13,100 lbs. per hour.

On the assumption of $H_1 = L$ we have therefore—

(1) $H_1 = 18,630,000$ B.Th.U.'s per hour.

(2) $H_1 = 13,550,000$ „ „

To arrive at the correct values for H_c , we must make some further assumptions, viz., that the loss of heat in the turbine due to radiation, etc., is 2 per cent of the heat converted into useful work, and that the condensate is drawn off at a temperature of 95° F. in both cases.

The following table illustrates the correct calculations for H_1 —

TABLE II.

Initial steam conditions	150 lbs dry sat	250 lbs. at 656.3° F.
Initial heat contents B.Th.U.'s per lb	1,201	1,352
Steam consumption. Lbs. per hour. (Q_s)	18,000	13,100
Total initial heat = H_2 . B.Th.U.'s	21,618,000	17,711,200
Work done $H_w = 1,500 \times 2,546$. B.Th.U.'s per hour	3,819,000	3,819,000
Heat lost through radiation, etc $H_r = 1\frac{1}{2}\% \times$ H_w . B.Th.U.'s per hour.	76,380	76,380
$H_c = Q_s \times (95 - 32)$. B.Th.U.'s per hour	1,134,000	825,300
$H_1 = H_2 - H_w - H_r - H_c$. B.Th.U.'s per hour	16,588,620	12,990,520
H_1 calculated from L. B.Th.U.'s per hour	18,630,000	13,558,500
Total difference	2,041,380	667,980
Error in assuming $H_1 = L$	approx. 11%	approx. 5%

It will be seen that in taking $H_1 = L$ regardless of the degree of wetness of the steam, we commit an error of fair magnitude. At the same time, since in doing so we are working with a margin, the error will not prejudice the resultant condensing plant adversely, and as a matter of convenience in condenser practice there is thus no great objection to doing so. The condenser designer more-over is usually given a certain steam quantity to work to, but he is rarely given any indication of the actual steam condition. He has therefore no alternative but to make his own assumption in this respect, and can only ensure being on the safe side by assuming the steam to be dry. It is, however, obvious that the actual heat contents require to be known or calculated in the above manner where accurate calculations are involved. The real objection to the practice of taking $H_1 = L$ lies in the fact that the error is a variable one, as in the case of our examples, and if a condenser

gives a certain result, and proves of ample capacity when working in conjunction with a certain prime mover, an identical plant may fail if it happens to be used with a prime mover whose exhaust steam has a higher heat contents.

Cooling Water Required. Isolated attempts have been made to dispose of the heat rejected in a condenser by passing air over the cooling surface, but the power consumption involved in attaining the necessary draught rules this method out of practical consideration except in locomotive practice.* The only effective means of providing the necessary cooling medium is by the direct heating of water, or, as in the case of condensers of the evaporative type, by evaporation of water in immediate contact with the outside atmosphere. The quantity of cooling water required is determined by the formula—

$$Q_w = \frac{H_1}{10 \times 60 \times (t_2 - t_1)}$$

Where t_1 = Inlet temperature of water in ° F.

t_2 = Outlet „ „ „

Q_w = Water quantity in gallons per minute

H_1 = Heat units to be absorbed per hour in B.Th.U.'s

$t_2 - t_1$ representing the rise in temperature of the water whilst passing through the condenser can be denoted by $D - d$, then

$$Q_w = \frac{H_1}{600 \times (D - d)}$$

This formula assumes unity specific heat for water at all temperatures which is not strictly accurate, but the error so introduced is negligible in practice. Where water contains a large percentage of impurities, its specific heat may be below unity—thus, sea water has a specific heat of about 0.94, so that a somewhat larger margin of cooling water should be allowed under such conditions than would otherwise be called for.

COOLING WATER TEMPERATURES. (The cooling water inlet temperature varies with the climate of the locality of the plant and is naturally dependent on the nature of the water supply available. In temperate climates where river or sea water is used it will vary from 32° F. in winter to 60° F. in summer. With cooling towers in use we can reckon on a variation from 70° to 80° F. The outlet temperature t_2 is necessarily below the temperature of the steam to

* A description of a purely air-cooled locomotive condenser appeared in *Engineering*, 1922, p 164, etc.

be condensed t_o , but should approach the latter as nearly as possible. In jet condensers $t_o - t_2 = d$ may easily be as low as 5°F. , and lower figures are frequently recorded. In surface condensers, since the water does not come into such intimate contact with the steam, we must reckon on a larger figure for d , and it is not usually advisable to estimate on a lower value than 8°F.

On the other hand, d should not exceed 12°F. under normal working conditions in land turbine practice. In marine practice much higher values for d are usual and vary up to $d = 50^\circ \text{F.}$, according to the type of vessel and the type of engine or turbine. Test results on surface condensers have been published from time to time, showing d as low as 5°F. ; for surface work it would be quite unsound to rely on so low a figure being obtained. We have therefore to make an arbitrary assumption for d to determine t_2 , t_o being taken as the temperature of evaporation for the vacuum in question.

Table III gives the quantity of cooling water required per 1,000 lbs. of steam in gallons per minute for varying vacua and inlet water temperatures for $d = 5$ to $d = 12$, based on $H_1 = L$, as well as the corresponding ratios between water and steam quantities. The latter are usually referred to as the number of times water required in relation to any given steam quantity.

(Table III will be found at the end of the book.)

CHAPTER IV

SURFACE CONDENSERS—GENERAL CALCULATION OF COOLING SURFACES

By far the largest number of condensers built to-day are of the surface type. For marine purposes they are now used practically exclusively, as the recovery of the condensate is essential. For large power land turbine installations also, the use of jet condensers is quite exceptional, as apart from the recovery of the condensate, the auxiliary power consumption of a jet condenser plant and the capital cost of the requisite air pumps to attain the extremely high vacua aimed at, represent serious drawbacks. For moderate size turbine units and for reciprocating engines, the jet type has still a large field of useful application, especially where an abundant supply of good feed water is available.

Principles of Design. In the most common form of surface condensers the water passes through the inside of the tubes and the steam enters the outer shell and impinges on the outer surface of the tubes. The reverse arrangement is occasionally adopted for pumping engines, when the whole body of the water to be handled by the pump is required to pass through the condenser with a minimum loss of head.

Most modern surface condensers are built on the counter-current principle, that is to say, the water path is opposed to the steam path, the water entering at the bottom of the condenser and leaving at the top, and the steam entering at the top and leaving in the shape of the condensate at the bottom.

The following constitute the leading features of a well-designed condenser. The tube surface and the size of the condenser should be a minimum consistent with an ample margin to compensate for the inevitable fouling of the tube surface.

The drop in steam pressure and the loss of water head should be minima. The quantity of circulating water required should not be excessive. Since these desiderata are partially contradictory, every design necessarily represents a compromise.

Calculation of Surface Area. The area of the cooling surface, Z sq. ft., required to condense a given weight of steam, Q , lbs. per hour, is determined from the law governing the transfer of heat from one substance to another, and through metals, which is practically analogous to Ohm's law for electric currents.

assumed. Newton's law of cooling upon which it is based deals only with the difference of temperature between the two bodies and remains unchanged under these circumstances. This fact is brought out more clearly by equation (10) where D is the temperature difference at the water inlet end and d that at the outlet end.

$$\text{From (10) we see } d_m = d \left[\frac{\frac{D}{d} - 1}{\log_e \frac{D}{d}} \right] \quad . \quad . \quad . \quad . \quad (12)$$

In this expression the factor within the bracket, which we shall call f , is a function of the ratio of $\frac{D}{d}$. The values for f for varying values of $\frac{D}{d}$ have been calculated and plotted in the curve shown in Fig. 4, and are also given in Table IV.

TABLE IV
FACTORS FOR CALCULATING MEAN TEMPERATURE
DIFFERENCES (LOGARITHMIC)

D/d	Factor f	Increment in f for increment of unity in D/d
1	1.000	
2	1.444	0.447
3	1.823	0.329
4	2.166	0.343
5	2.488	0.322
6	2.793	0.305
7	3.004	0.291
8	3.365	0.281
9	3.638	0.273
10	3.905	0.267

With the use of this curve, the value of d_m can be determined for any usual ratio of $\frac{D}{d}$ by simple slide rule method without the use of logarithmic tables.

Thus, if $t_o = 105^\circ \text{ F.}$

$$t_1 = 60^\circ \text{ F.}$$

$$t_2 = 96^\circ \text{ F.}$$

then $D = 45$ and $d = 9$ and

$$\frac{D}{d} = 5.$$

From the curve we find $f = 2.49$, then $d_m = 9 \times 2.49 = 22.41$.

APPROXIMATION. For the purpose of arriving at an approximation of Z we can assume d_m to be the arithmetic mean between $t_o - t_1$ and $t_o - t_2$

$$d_m = \frac{D + d}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

CHAPTER V

SURFACE CONDENSERS—DETERMINATION OF THE HEAT TRANSMISSION COEFFICIENT K

As a general rule, designers find it convenient to use an empirical figure for K, varying according to conditions between limits of 200 and 800 B.Th.U.'s per square foot per ° F. per hour in turbine practice, but lower values are used in condensers working in conjunction with reciprocating engines. A number of writers on the subject have attempted to develop formulae, which in conjunction with experiments carried out by themselves or published from time to time, would enable the designer to predetermine K with a fair degree of certainty. Unfortunately it has been found, so far, that if any such formulae are compared with published test results observed under careful conditions they fail to show consistent results. The difficulty really lies in the fact that so many factors enter into the calculation, and that the effect of each factor is exceedingly difficult to trace by experiment. Even where the effect of an individual factor can be said to be clearly determined, the question immediately arises how to co-ordinate the result in conjunction with the variations of the remaining factors which enter into the problem. The present authors do not therefore consider it advisable or even practicable to attempt to develop a rational formula for the determination of K, but an attempt will be made in the following notes to explain as far as possible the manner in which K is affected by variations in conditions, since an appreciation of the whole of the conditions controlling the transfer of heat from steam to cooling water is of fundamental importance to the designer.

Subdivision of R and K. The total resistance to be overcome by the heat in passing from steam to cooling water may be considered to be made up of a number of resistances in series thus—

$$R = R_1 + R_2 + R_3$$

where R_1 = resistance between steam and the outside surface of the tubes, R_2 = resistance across the metal, R_3 = resistance from the inside surface of the tubes to the cooling water, or if K, K_1 , K_2 , K_3 represent the reciprocals or the corresponding heat transmission coefficients, then—

$$\frac{1}{K} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3}$$

If we consider unit area, the temperature fall is given in each case by the relation

$$d_{m1} = \frac{\alpha}{K_1}$$

$$d_{m2} = \frac{\alpha}{K_2}$$

$$d_{m3} = \frac{\alpha}{K_3}$$

where d_{m1} = mean temperature difference between steam and outer surface of tubes.

„ d_{m2} = mean temperature difference across the tube.

„ d_{m3} = mean temperature difference between inside surface of the tube and the cooling water.

α = B.Th.U.'s transferred per square foot per hour and
 $d_{m1} + d_{m2} + d_{m3} = d_m$.

Value of R_2 . The heat transmission coefficients for metals are well known, and for brass tubes we can assume 755 B.Th.U.'s per square foot per hour per degree F. per inch thickness. For a tube No. 18 B.W.G. (0.049 ins. thick) K_2 = approximately 15,000,

$$\text{or } R_2 = \frac{1}{15,000}$$

This resistance forms such a small proportion of the total resistance as to be entirely negligible in the calculation. There is a certain irony in the fact that the one factor which can be predetermined with any degree of certainty should be of negligible magnitude so that the determination of this quantity does not help us materially towards the solution of the problem. If scale or vegetable slime is deposited by the cooling water on the inside of the tubes, we have to deal with a material having a very low heat transmission coefficient and which, of course, varies with the nature of the deposit. If the scale is of the most common lime and magnesia composition, then the heat transmission coefficient would probably be similar to that usually quoted for boiler scale, that is of the order of 15 B.Th.U.'s per degree F. per hour per inch thickness, so that a coating of $\frac{1}{32}$ in. thickness would interpose a resistance $R_{2 \text{ scale}} = \frac{1}{480}$

EFFECT OF INCRUSTATION ON HEAT TRANSMISSION. If the condenser is designed to perform its normal duty with clean tubes on the

basis of a total heat transmission coefficient of $K=300$, then the formation of $\frac{1}{32}$ in. of scale will increase the resistance to heat flow in the ratio of $\frac{1}{300} : \frac{1}{300} + \frac{1}{480}$, which is equivalent to a reduction in capacity of 38.5 per cent.

If on the other hand the condenser is designed on the basis of $K = 600$, then the resistance increase will be $\frac{1}{600} : \frac{1}{600} + \frac{1}{480}$, or a capacity reduction of 55.5 per cent.

We can, therefore, conclude that the formation of scale has a more pronounced effect on a condenser designed with a high heat transmission coefficient than one designed with a low coefficient, always providing that the same proportion of cooling surface is devoted to air cooling in both cases. On the other hand, this disability of the condenser, based on a high K is partly off-set by the fact that the formation of scale, and to a greater extent that of a layer of slime, is less likely to occur where the water velocity is high, a condition always associated with a high K , as will be shown later.

It follows that if the designer has to contend with cooling water that is hard or that contains a considerable amount of vegetable matter or similar substances, it is advisable to protect the plant against rapid deterioration of its performance by allowing a margin on the cooling surface, that is by working with a relatively low figure for K , whilst at the same time maintaining the water velocity at a comparatively high figure.

Determination of R_1 . As steam condenses on a tube, a film of water is formed completely coating the tube so that, except for the short initial period of steam admission, the steam does not come into immediate contact with the metallic surface, and the heat has to overcome an additional resistance due to the water film. Consequently, where values of R_1 are determined by experiments, such values must be taken to include the resistance of the water film.

Professor Callendar reported the results of a series of experiments which he carried out at McGill University,* for the purpose of ascertaining the effect of the variations in the thickness of the water film. Quite contrary to his expectations and to the general

* *Report of 67th Meeting of the British Association at Toronto, 1897.* (John Murray.)

assumption, these experiments indicated no appreciable improvement in the rate of heat transfer when the water film was reduced. The conditions of the test were, however, very different from those generally met with in condenser practice, as they were carried out primarily in connection with the determination of cylinder condensation losses, more especially in that a rapid downward flow of water was maintained.

Professor Callendar has expressed an opinion that the apparent small effect of the thickness of the water film is due in these experiments to—

- (1) The rapid downward flow of the condensate.
- (2) The high temperatures at which the experiments were made (250° to 300° F.) on account of their relation to engine investigations and the resulting low viscosity of the condensate.
- (3) The film of condensate in contact with the tube is not quiescent but is in rapid transverse motion, the particles of water acting as conveyors of heat, the transmission of heat thus not being dependent solely upon conduction.
- (4) The steam used was practically free from air, a condition which does not obtain in condenser practice.

In the usual type of condenser, the conditions are, of course, entirely different. Since the tubes are placed horizontally, the velocity of the flow of the condensate over the tube surface would be less, and the conductivity of the water would be further reduced on account of the higher viscosity at vacuum temperatures, so that it is reasonable to assume that the thickness of the water film does, in fact, affect R_1 . This is borne out by experiments carried out by Nusselt, quoted by Professor A. Stodola,* where two rows of tubes were arranged vertically over each other, so that the condensate formed on the upper row dropped immediately on to the lower row, and the rate of heat transfer was reduced in the lower row to 0.68 of that obtained in the upper row. This constitutes one reason why it is desirable to avoid, as far as this may be practicable, building condensers of too great a depth. A more important reason is the necessity of reducing to a minimum the drop of pressure across the condenser and therefore the length of the path through the condenser. This factor is of lesser moment in small condensers but attains pronounced importance in large units, and a number of designs illustrated in Chapter X show the methods adopted by various manufacturers for dealing with this problem.

EFFECT OF PRESENCE OF AIR IN CONDENSERS. The air carried over with the steam remains in contact with the tubes after the

* Stodola *Die Dampfturbine*. (Fifth Edition, p. 737.)

steam has been condensed and, if allowed to accumulate, it will gradually blanket complete sections of the condenser and greatly impair its efficiency. This effect was clearly shown in the course of experiments carried out by Dr. Bengough for the Committee on Corrosion of Condenser Tubes of the Institute of Metals,* which were carried out on a condenser having a small steam opening in the centre. Where the air accumulates, the partial steam pressure and consequently the temperature of the mixture of steam and air falls. Dr. Bengough placed thermometers at either end of and in the centre of the tubes opposite the steam inlet. The lower readings at either end showed that the air was more or less stagnant and accumulating at these points, whereas it was being swept away by the rush of steam at the centre. The aim of all modern condenser designers is to ensure that the steam velocity in all parts of the condenser shall be maintained at a sufficiently high velocity to prevent the accumulation of air and to keep the air in motion from steam inlet to air pump suction. This is seen in the formation of steam lanes by suitable spacing of the tubes, and by the wedge and pear-shaped condenser bodies, and above all by the provision of very ample steam openings.

NUMERICAL VALUE OF R_1 There does not appear to be any available experimental data which would form a conclusive basis for calculating R_1 . Professor Josse† expresses the opinion that the rate of heat transmission from steam to tubes K_1 in British units of approximately 4,000, determined by Ser, is not in any way too high for modern condenser designs, and concludes from this that very little can be gained by attempting to introduce improvements in design for the purpose of reducing the value of R_1 . This conclusion seems quite sound providing that precautions are taken to prevent the accumulation of air on the lines already indicated.

Determination of R_3 . There remains the calculation or determination of R_3 the resistance from the inside of the tube to the cooling water, and since this resistance is usually the greatest of the three, its magnitude practically determines the value of the whole.

The dependence of R_3 on the water velocity has long been generally recognized, but as Mr. A. E. Leigh Scanes pointed out in a paper on "Modern Surface Condensing Plants," which he read before the Institute of Mechanical Engineers,‡ a summary of experiments

* *Engineering*, 5th September, 1913 (p 339)

† *Mitteilungen aus dem Maschinen Laboratorium der Technischen Hochschule, Berlin*, 1913 (R Oldenbourg, Berlin)

‡ *Proc Inst Mech Eng*, 1913 (Parts I and II, pp 179-252)

carried out by ten different observers to determine the effect of the water velocity on R_3 showed most divergent results. The experiments all agree in that they indicate a substantial fall in R_3 with rising water velocity, but the actual results do not coincide in any two instances. Thus, whereas Joule's experiments give $K_3 = 525$ for $V = 5$ ft. per sec., those due to Orrok give $K_3 = 690$ for the same water velocity, whereas Ser and Josse arrive at this value for K at water speeds of approximate 2.5 and 3 ft. per sec.

CRITICAL WATER VELOCITY. In an article in *Engineering*,* Mr. H. M. Martin draws attention to the fact that the experimenters have failed to take full account of the theoretical considerations initially expounded by Osborne Reynolds. In a paper read before the Royal Society in 1883,† Reynolds explained the following experiments.

Water was allowed to flow at varying velocities through a glass tube submerged in a glass tank. This tube was bell-mouthed and highly coloured water could be released through a small orifice opposite to the bell mouth. This coloured water was drawn into the tube, and up to a certain velocity it was extended into a straight line right through the tube. As the velocity was increased by opening a control cock on the discharge, a water speed was reached when the thin line broke up completely and the coloured water spread over the whole diameter of the tube. A careful examination of the experimental results enabled Reynolds to confirm deductions made from the equations of motion, and he ascertained that the flow of water in tubes changed abruptly from viscous to turbulent, the critical velocity in m. per second at which the change takes

place being
$$V_{\text{critical}} = \frac{P}{BD}$$

where P = Poiseuille's ratio of viscosity to density

$$= \frac{1}{1 + 0.03368T + 0.000221T^2}$$

D = Internal diameter of tube in metres.

B = Constant = 43.79; and

T = Temperature in degrees centigrade.

So long as the water is flowing in viscous or stream-line motion, that is to say, in annular parallel layers, the heat travels by conduction from the outermost layers to the central core, and since the

* *Engineering*, 1914 (p. 38).

† See also *Scientific Papers* (Vol. II, p. 51) on the "Motion of Water and of the Case of Resistance in Parallel Channels."

individual layers retain their relative radial position the hottest layer remains in contact with the tube, and the temperature difference between layers and consequently the rate of heat transfer is a minimum. As soon as turbulence is established the stream line motion is destroyed and the water obtains a radial motion, with the result that particles of water that are at one moment at the core and, consequently, at the greatest distance from the source of heat, and at the minimum temperature but with the maximum axial

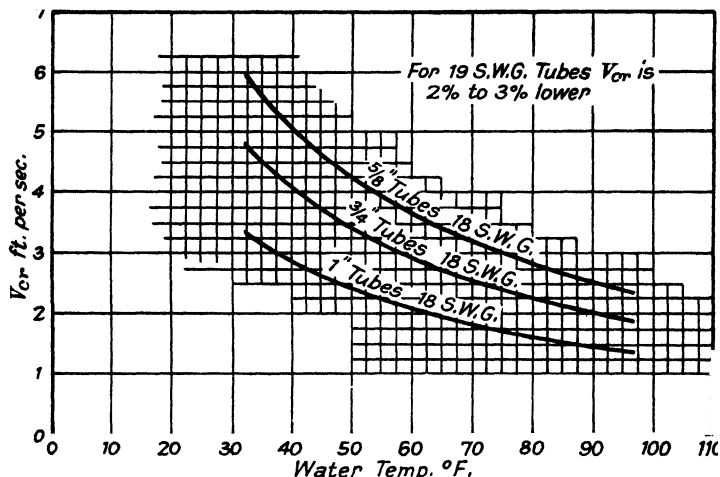


FIG. 5.

CRITICAL WATER VELOCITY FOR VARYING WATER TEMPERATURES AND TUBE SIZES.

velocity, move out to the periphery and, therefore, a greater temperature difference and a higher rate of heat transfer is established between the tube surface and the outer particles of water. Further, the mixing action in the water itself provides a better opportunity for the heat to become evenly distributed over the whole area than is obtained by conduction across layers, as the particles of water act as conveyors of heat to the core, whilst on the return journey to the walls they act as conveyors of velocity, thus averaging the velocity across a tube.

The curves shown in Fig. 5 show the critical water velocities determined from Reynolds' formula for varying inlet water temperatures, and for 1 in., $\frac{3}{4}$ in. and $\frac{5}{8}$ in. tubes, No. 18 S.W.G.

SUITABLE VELOCITY FOR FLOW OF WATER. Osborne Reynolds' experiments indicated that the water is in an unstable condition

before it reaches the critical velocity, and the viscous condition of flow is most probably easily upset at lower velocities. Since the water path is by no means an even one at the entry to a condenser and in passing from the water box to the tubes, it seems doubtful whether the flow in the tubes would be viscous even at velocities below the critical. Lastly, we have to consider that water entering a tube at a velocity below the critical may become turbulent by virtue of an increase in temperature. If, therefore, the initial water conditions at the water inlet fall below the critical, it does not by any means follow that this condition will obtain throughout the condenser, but the designer can generally avoid having to provide the high resistance R_3 inherent to water in viscous flow by choosing a water velocity above the critical for the specified inlet temperature and tube diameter.

VISCOUS ANNULUS IN TURBULENT FLOW. Where the water flow is entirely viscous the velocity varies from a maximum at the axis to zero at the point of contact with the tube. If r_o is the inside radius of the tube, then the velocity at any radius is given by the formula

$$V = \frac{p}{4\mu} (r_o^2 - r^2)$$

where p = pressure drop of water per unit length of tube.

μ = viscosity of water, and the average velocity is given by

$$V = \frac{pr_o^2}{8\mu}$$

so that in viscous flow the average velocity is half the maximum velocity and the velocity curve is represented by a parabola.

If the velocity is increased until turbulence sets in, then there will still remain an outer annulus in viscous flow which interposes an additional resistance.

The probability of the presence of this annulus of water in viscous flow under conditions when the bulk of the water is in turbulent flow, had been suspected for some time before its existence was established by experiments due to Dr. Stanton.*

In view of the relatively high resistance due to even a thin layer of water in viscous flow, this factor must enter largely into any attempt to calculate R_3 , and it becomes necessary to subdivide R_3 .

It would therefore appear that R_3 is composed of two more or less distinct components R_{3v} , the resistance due to the viscous film

* *Engineering*, 1921, 20th July (p. 10).

Inserting in this the value of $\frac{dt}{dx}$ from (2) we then get, after simplifying

$$q = (T - t) \frac{F}{v} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

But the heat transfer per unit area is equal to the transmission coefficient (K_s) multiplied by the mean temperature difference ($T - t$) or using the reciprocal of K_s (i.e. the resistance to heat flow R_s)

$$q = \frac{T - t}{R_s}$$

Hence substituting in (4) we are left with

$$R_s = \frac{v}{F} \text{ or } K_s = \frac{F}{v} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

This shows that the heat transmission coefficient varies directly as the frictional resistance to flow and apparently inversely as the velocity. However, F varies with the velocity as according to Stanton & Pannell

$$F = \rho v^2 L$$

where L is Lee's factor and is a function of $\frac{\mu}{\rho v d}$, μ being the viscosity of the fluid.

Inserting this value in equation (2)

$$T - t = \frac{d}{4L} \frac{dt}{dx}$$

CONSIDERING THE FLOW AS ENTIRELY VISCOUS. Martin then refers to the evidence that in turbulent flow through a tube there is a film in viscous flow adhering to the tube.

Consider initially that the flow is entirely viscous and examine the flow through a ring of radius r , thickness Δr , length Δx . Due to viscosity the force acting on the fluid in this space at the inner surface is

$$F = 2\pi r \mu \frac{dv}{dr}$$

At the outer surface the force is

$$F + \frac{dF}{dr} \Delta r,$$

and so the resultant tangential force is

$$\frac{dF}{dr} \Delta r = 2\pi \mu \frac{d}{dr} \left(r \frac{dv}{dr} \right) \Delta r$$

The axial pressure at one end of the space is $2\pi r p dr$ when p is the fluid pressure. At the other end of the space the axial pressure is

$$2\pi r dr \left(p - \frac{dp}{dx} \Delta x \right)$$

so that the whole force acting on the fluid within the space is

$$2\pi \mu \frac{d}{dr} \left(r \frac{dv}{dr} \right) \Delta r - 2\pi r dr \frac{dp}{dx} \Delta x$$

Equating this force to the product of the mass flow by acceleration and simplifying we get

$$\frac{d}{dr} \left(r \frac{dv}{dr} \right) - \frac{r}{\mu} \frac{dp}{dx} = \frac{rm}{\mu} \frac{dv}{dt}$$

If the motion is steady, as is usually the case, $\frac{dv}{dt} = 0$. Further, it is sufficiently accurate to assume that the pressure drop is uniform along the tube so that $\frac{dp}{dx} = \frac{p}{l}$ where p = pressure drop in length of tube l .

$$\text{Hence } \frac{d}{dr} \left(r \frac{dv}{dr} \right) - \frac{r}{\mu} \cdot \frac{p}{l} = 0$$

Assuming that $r \frac{dv}{dr} = b$, substituting and integrating we eventually get as a solution

$$v = \frac{p}{4\mu l} (r_o^2 - r^2)$$

where r_o = outer radius of tube.

Considering the axial equilibrium of the whole section of the tube we would get

$$- \frac{\pi}{4} d^2 \frac{dp}{dx} \Delta x + F \pi d \Delta x = 0$$

$$\text{or } \frac{dp}{dx} = \frac{4F}{d}$$

$$\text{Hence } v = \frac{F}{\mu d} (r_o^2 - r^2) = a (r_o^2 - r^2). \quad . \quad . \quad . \quad (6)$$

TAKING THE FLOW AS TURBULENT. Imagine that the velocity is such that the core is in turbulent flow. A thin film of the water near to the walls of the tube will, however, remain in viscous flow, a point which was overlooked by Osborne Reynolds. Martin

states that the film in contact with the walls is, unchanged as regards its velocity, etc., and he assumes that the mean velocity of the core is equal to the maximum velocity in the film.

The flow in unit time through the viscous film is obtained thus from equation (6), r_o is the inner radius of the tube, and r_f the inner radius of the film, the thickness of the film being f .

$$\begin{aligned}\text{Volume of flow } V_f &= 2\pi a \int_{r_f}^{r_o} r (r_o^2 - r^2) dr \\ &= 2\pi a \left(\frac{r_o^4}{4} - \frac{r_o^2 r_f^2}{2} \right) + \frac{r_f^4}{4}\end{aligned}$$

From Martin's assumption regarding the mean velocity of the core

$$\begin{aligned}\text{Volume of flow in core } V_c &= \pi r_f^2 \cdot a(r_o^2 - r_f^2) \\ &= 2\pi a \left(\frac{r_o^2 r_f^2}{2} - \frac{r_f^4}{2} \right)\end{aligned}$$

$$\begin{aligned}\text{So that the total flow } V &= 2\pi a \left(\frac{r_o^4}{4} - \frac{r_f^4}{4} \right) \\ &= \frac{\pi F}{\mu r} \left(\frac{r_o^4}{4} - \frac{r_f^4}{4} \right)\end{aligned}$$

$$\text{Actually the total flow } V = v \times \pi r^2$$

$$\text{so that } \frac{r_f}{r_o} = \sqrt[4]{1 - \frac{4\mu v}{Fr_o}}$$

Substituting for F

$$\frac{r_f}{r_o} = \sqrt[4]{1 - \frac{4\mu}{\rho v L r_o}}$$

This enables the thickness of the film to be calculated, thus

$$\begin{aligned}f &= r_o \left(1 - \frac{r_f}{r_o} \right) \\ &= r_o \left(1 - \sqrt[4]{1 - \frac{4\mu v}{r_o F}} \right)\end{aligned}$$

$$\text{As } f \text{ is small in relation to } r_o, f = \frac{\mu v}{F}$$

This is based on an assumption regarding the maximum core velocity for which there does not appear to be any evidence quoted.

This film of thickness f transmits heat by conduction, and the resistance to the flow R_f is given by

$$R_f = \frac{f}{c} = \frac{\mu v}{cF}$$

c being the specific conductivity of the fluid.

In calculating R as given by equation (5), it was assumed that the core came into contact with the wall of the tube, but it only extends to the inner radius of the film. This involves a modification to the result. In practice the film is not as thick as the ideal film, and if the ratio between the actual and the ideal be φ , Martin shows by reasoning similar to that used earlier that $R_c = (1 - \varphi) \frac{v}{F}$ and also $R_f = \frac{\varphi f}{c}$

The total resistance to the transfer of heat is

$$R = R_f + R_c, \text{ so that } R = \frac{v}{F} \left(\frac{\varphi u}{c} + 1 - \varphi \right)$$

Having obtained this expression for the resistance to heat transmission, Martin then considers various experiments on condensers, conferring his attention mainly to those of Webster.* He ultimately arrives at a table giving the core resistance R_c , the film resistance R_f , the total resistance ($R_c + R_f$) and the tube internal surface temperature for a series of widely spaced conditions.

It remains to be proved whether Martin's tables can be taken as a definite basis for the calculation of R_2 . If further experiments confirm the accuracy of these tables, then it will be desirable to reduce them to simpler form, even if this involves some sacrifice of accuracy, as they are very laborious to apply to condenser calculations in their present form.

In any case, there can be no doubt that the water velocity enters very largely into the calculation, and every attempt to pre-determine the surface area required to perform a given duty, and all comparisons of test results obtained on surface condensers, should allow for this important factor.

CHAPTER VI

SURFACE CONDENSERS—COOLING SURFACE FOR AIR AND CONDENSATE

WE have so far only considered the surface area required for condensing the steam. The condenser has, however, to fulfil the additional functions of cooling the entrained air and of cooling the condensate, and a proportion of the total surface must be allocated to these functions.

The necessity for reducing the air temperature arises from the following consideration.

If a continuous flow of steam free from air entered the condenser, then if we neglect the drop in pressure required to produce the steam flow, the process of condensation would take place throughout the condenser at the same pressure and temperature, and the cooling effect would be solely utilized in extracting the latent heat from the steam, apart from the cooling of the resultant water already referred to which would take place, due to particles of water when formed remaining in contact with the cooling surface for some time before forming into drops and falling away from the tubes. Neglecting this cooling effect on the water, the condensate would leave the condenser, at the same temperature as the steam enters the condenser providing the steam is not superheated at this stage. Since in practice the steam is not free from air, the conditions of condensation are altered.

Effect of Presence of Air on Temperature of Condensation. If a vessel contains water in contact with a gas then steam or vapour is formed, and the steam pressure is equal to that of saturated steam at the prevailing temperature. The space in a condenser that is not occupied by the condensate will therefore contain a mixture of steam and air, and the steam pressure at any point will be equal to that of saturated steam at the temperature of the mixture. This condition applies at all stages of the condenser from the steam inlet to the air outlet. It is therefore, of course, inaccurate to differentiate as we do between a steam inlet and an air outlet, since the gas entering the condenser as well as that leaving the condenser consists of a mixture of steam and air and of any other gases that may be present.

The essential difference as between the points of inlet and outlet lies in the ratio of steam quantity to air quantity.

We shall consider the problem on the basis of Dalton's law, although this law has not been proved to apply to gases in motion.

It is reasonable to assume that the temperature of the steam and that of the air is the same at all points in the condenser. According to Dalton's law, the total pressure due to a mixture of any two gases is equal to the sum of the pressures of the individual gases, that is

$$P = P_s + P_a \quad (1)$$

and the volume of each gas is equal to the total volume

$$V = V_s = V_a \quad (2)$$

P_s and P_a are usually referred to as the partial steam pressure and the partial air pressure. It follows that the actual pressure at the steam inlet, that is the pressure recorded by the vacuum gauge at this point, is not equal to the steam pressure but represents the sum of the steam and air pressures,

$$P_e = P_{se} + P_{ae} \quad (3)$$

The steam temperature t_o and the steam volume are those corresponding to saturated steam of a pressure of P_{se} , providing again that the steam is not superheated.

Assuming that the total pressure P_e at the air pump suction is equal to that at the condenser inlet—in fact, the pressure must be slightly lower to produce the necessary flow of gases—then since the steam volume has been reduced by condensation, the air volume must also be reduced, and this can only be due to cooling of the air and to raising of the partial air pressure which, in turn, involves a reduction of the partial steam pressure which can only take place if the steam temperature is reduced.

The Necessary Drop in Temperature. If the temperature were to remain constant, then the ratios between air volume and pressure and steam volume and pressure would be unchanged, and the air pump would have to withdraw air and steam in the same proportions as they entered the condenser. That is to say, unless the air were allowed to accumulate in the condenser and destroy the vacuum, the whole of the steam would have to be withdrawn by the air pump—an obviously impossible condition. It follows from these considerations that, where air is present in the condenser, condensation cannot be effected at constant steam pressure and temperature, but that the temperature of the mixture must be reduced and surface area must be provided to cool the mixture.

The amount of the reduction is dictated by the amount of steam which may be carried over into the air pump. Where the air and

condensate are withdrawn by the same pump this is not a serious consideration, as the steam so carried over is mainly condensed in the air pump, due to the rise in pressure and the cooling of the mixture by the condensate, and the amount of steam carried away by the air is then negligible.

Where the air is withdrawn separately from the condensate, this quantity of steam represents a loss of feed water.

Calculation of Steam Quantity Carried Over by the Air Pump. To arrive at the amount of the temperature reduction required to reduce the steam quantity carried over to a given figure, we require to know the amount of air, Q_a , entering the condenser. We can then determine the steam quantity carried over in the following manner.

If the specific volume of the steam corresponding to the partial steam pressure P_{se} is v_{se} , then since the total air volume is equal to the total steam volume the specific volume of the air is—

$$v_{ae} = \frac{Q_s \times v_{se}}{Q_a} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The air pressure P_{ae} is determined from the well-known formula for gases—

$$\frac{Pv}{T} = R \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Where P = pressure of air in lbs. per sq. ft.

T = absolute temperature in ° F.

v = volume of gas in cu. ft. per lb.

R = 53.18 for air

$$\text{or } P_{ae} \text{ in lbs. per sq. in.} = \frac{t_a + 459.6}{2.7093 \times v_{ae}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The total pressure at the condenser inlet is then determined as the sum of partial steam and air pressures. If now the temperature of the mixture at the air pump suction is t_c , then the corresponding partial steam pressure P_{sc} is determined from the steam tables and the air pressure—

$$P_{ac} = P_c - P_{sc} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$\text{where } P_c = P_e - \Delta P \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

ΔP being the drop in total pressure between condenser inlet and air pump inlet.

$$\text{Then } v_{ac} = \frac{t_c + 459.6}{2.7093 \times P_{ac}} \quad (9)$$

$$\text{and } V_{ac} = v_{ac} \times Q_a \quad (10)$$

$$\text{and } V_{sc} = V_{ac} \quad (11)$$

If v_{sc} is again obtained from the steam tables, then

$$Q_{sc} = \frac{V_{sc}}{v_{sc}} \quad (12)$$

If a number of values for Q_{sc} are determined in this manner for various values of t_c , and a curve plotted showing Q_{sc} as a function of t_c , then the necessary drop in temperature for any given value for Q_{sc} can be determined.

EXAMPLE. If we consider a condenser dealing with

$Q_s = 10,000$ lbs. of steam per hour, and

$Q_a = 10$ lbs. of air per hour, then if the inlet temperature

$t_o = 100^\circ$ F. and

$P_{se} = 0.95$ lb.

$v_s = 350$ cu. ft. per lb.

$V_{ae} = V_{se} = 3,500,000$ cu. ft., and

$v_{ae} = 350,000$

$$P_{ac} = \frac{100 + 459.6}{2.7309 \times 350,000} = 0.000593 \text{ lb. per sq. in.}$$

$P_e = 0.9506$ lbs.

Assuming that no drop in pressure occurs across the condenser, that is $P_c = P_e$ and taking t_c at 90° F., we find from steam tables

$P_{sc} = 0.7$ and $v_{sc} = 466$

$P_{ac} = 0.9506 - 0.7 = 0.2506$

$$v_{ac} = \frac{90 + 459.6}{2.709 \times 0.2506} = 812 \text{ and } V_{ac} = 8120, \text{ and}$$

$$Q_{sc} = \frac{8120}{466} = 17.4 \text{ lb. per hour.}$$

Therefore $Q_{sc} = 0.174$ per cent of Q_s .

The curves shown in Fig. 6 show the calculated values of Q_{sc} as a function of t_c for $P_c = P_e$ and again for $P_c - P_e = 0.1$ lb. It will be seen that if it is intended to limit the loss of steam or of feed water to 0.25 per cent, then the required drop in temperature is 7.5° F. in the case of $P_c = P_e$, and 11.5° F. where $\Delta P = 0.1$ lb. under the conditions assumed for the present example.

It will be seen that where, as in the case of the present example, Q_a is small in comparison with Q_s , the difference between P_{se} and P_e is too small to have any appreciable effect on the resultant value of Q_{sc} , so that the calculation can be simplified by leaving out the initial calculation of P_e and using $P_{se} - \Delta P$ in (7) in place of P_e or $P_e - \Delta P$. Thus,

$$P_{ac} = P_{se} - \Delta P - P_{sc} \quad (7a).$$

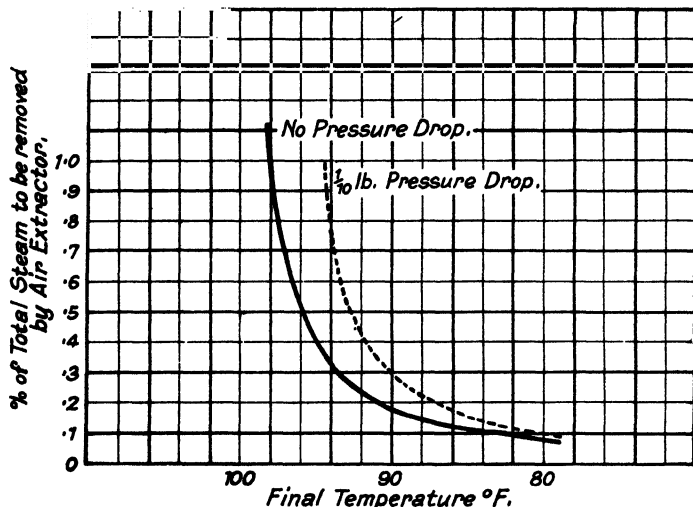


FIG. 6
STEAM QUANTITY CARRIED OVER TO AIR PUMP.

We can also neglect the variation of t_c in formula (9) and take $t_c + 459.6$ as a constant = 560.

$$v_{ac} = \frac{560}{2.709 \times P_{ac}} = \frac{207}{P_{ac}} \quad (9a).$$

If we aim at calculating Q_{sc} as a percentage of Q_s , then

$$Q_{sc} = \frac{Q_a \times 100 \times 207}{Q_s \times P_{ac} \times v_{ac}} \text{ per cent} \quad (13)$$

$$\text{or if } \gamma = \frac{Q_a}{Q_s} \quad (14)$$

$$Q_{sc} = \frac{20,700 \gamma}{P_{ac} \times v_{ac}} = \frac{20,700 \gamma}{(P_{se} - \Delta P - P_{sc})} \times v_{sc} \text{ per cent} \quad (15)$$

The calculation of Q_{sc} as a percentage of Q_s , and the construction of the curve $Q_{sc} t_c$ for any given vacuum, and drop across the

condenser for any given ratio of air to steam quantity, only involves the use of formula (15), in which P_{sc} and v_{sc} are the only variables and can be read from steam tables for any value of t_c . If we apply formula (15) to our previous example then we have—

$$\gamma = \frac{1}{1000} \quad P_{se} = 0.95, \quad \Delta P = 0, \quad P_{sc} = 0.7, \quad \text{and} \quad v_{sc} = 466.$$

$$Q_{sc} = \frac{1}{\frac{20,700 \times 1000}{(0.95 - 0.7) \times 466}} = 0.178 \text{ per cent against the corresponding}$$

figure 0.174 per cent, arrived at by means of the more accurate calculation, and it is seen that difference is entirely negligible for our purpose.

Heat Transmission Coefficient for Air. If the specific heat of air is taken as 0.2375, which is Regnault's figure for air at constant volume and for a temperature range of 0 to 200° C., and the temperature reduction to be obtained be 10° F., which should be ample for most conditions to reduce the loss of feed water to a negligible quantity where the plant is reasonably air-tight, then the total amount of heat to be transferred from the air to the cooling water $H_a = 10 \times 0.2375 = 2.375$ B.Th.U.'s per lb. of air handled. If, as in our last example, 1 lb. of air is admitted per 1,000 lbs. of steam, then the corresponding amount of heat transferred from the steam is approximately $1000 \times 1000 = 1,000,000$ B.Th.U.'s. If, therefore, the heat transmission coefficient from air to water K_a were of the order of even one thousandth of that from steam to water K , then the amount of cooling surface required to effect the cooling of the air would be of negligible magnitude even where a far higher drop in temperature is aimed at than that here assumed.

The effect of the presence of air in a condenser in a practically stationary condition, in so far as it blankets tube sections and hinders the contact of the steam with the outer tube surfaces, has already been considered. Many authenticated observations are available from readings taken under ordinary commercial conditions where the rise in water temperature across the first or bottom pass was exceedingly low, that is to say the corresponding cooling surface was inactive in condensing steam, and it is reasonable to assume that this condition was not due to the blanketing effect referred to, as this would not be likely to be confined to the bottom part of the condenser, but rather to the apparently inactive surface being engaged in cooling the air. This condition is most frequently met with where an excessive amount of air is present, or where, owing to insufficient volumetric capacity of the air pump, the air has to be cooled below

the degree required for the purpose of reducing the steam quantity carried over. Since it is, therefore, apparent that under certain conditions an appreciable proportion of the cooling surface may be occupied in cooling the air, it follows that K_a must be of an exceedingly low order.

Mean Temperature Difference. From Fig. 7 it will be seen that the bulk of the steam is condensed with a very small drop in temperature. Thus, under the conditions chosen for curve 1, 99 per cent of the steam is condensed with a drop in temperature of 1°F. , so that if we neglect the amount of air cooling which is accompanied by the condensation of the bulk of the steam quantity, then the rise in temperature of the cooling water due to air cooling is negligible, and we can calculate the available mean temperature difference on the basis of constant cooling water temperature, namely, the water inlet temperature t_1 . If the air temperature is to be reduced from t_{ea} or, say, t_o to t_{ca} ,

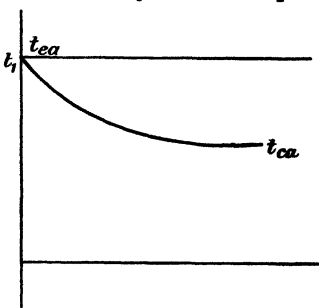


FIG. 7.
MEAN TEMPERATURE DIFFERENCE FOR AIR.

$$\text{then } d_{ma} = \frac{t_o - t_{ca}}{2} - t_1,$$

or, more accurately, if $t_{ea} - t_1 = D_a$ and $t_{ca} - t_1 = d_a$,

$$\text{then } d_{ma} = \frac{D_a - d_a}{\log_e \frac{D_a}{d_a}} \text{ and } Z_a = \frac{H_a}{K_a \cdot d_{ma}}$$

where Z_a = surface to be allowed for air cooling.

The experimental data and formulae due to various authorities which have been published for the determination of K_a are generally only applicable to conditions which do not prevail in a condenser. It appears to be established that K_a varies in an inverse degree with the pressure, and where the air velocity is above the critical, K_a no doubt increases with the velocity. In a condenser the air velocity in relation to the whole tube area is generally a variable quantity, and of a value which it would hardly be practicable accurately to determine. It is probable that certain particles of the air do not come into direct contact with the tubes at all, and these particles can therefore only be cooled by conduction across

other particles of air at a considerably reduced rate, as air is a notoriously bad conductor of heat. It is therefore obvious that it is impracticable to predetermine K_a with any degree of accuracy. In any case it is reasonable to assume that the resistance from air to the metal is so high in comparison with that across the metal or from the metal to the water that these other two resistances can be ignored, and we can say $R_a = R_{a1}$ or $K_a = K_{a1}$.

Surface Allowed for Air Cooling. In an article on the calculation of surface condensers, Dr. Hoefer* bases his estimate of Z_a on a value of $K_a = 0.0104$, derived from certain experiments carried out by Professor Josse.

If in our previous example $t_1 = 60^\circ \text{ F.}$ and $t_2 = 90^\circ \text{ F.}$ and $K = 400$,

$$\text{then } d_m = -\frac{30}{\log_e \frac{40}{10}} = \frac{30}{1.386} = 21.6$$

$$\text{and } Z = \frac{10,000 \times 1034}{400 \times 21.6} = 1200 \text{ sq. ft.}$$

Taking Dr. Hoefer's value for K_a , and if $t_{ea} = 90^\circ \text{ F.}$, then approximately

$$d_{ma} = \frac{190}{2} - 60 = 35$$

$$Z_a = \frac{2.375}{0.0104 \times 35} = 6.8 \text{ sq. ft.}$$

If Dr. Hoefer's value is representative of the value of K_a to be expected under average conditions, then it would appear from these figures that the amount of air cooling needed to meet all general requirements in the matter of loss of feed, due to vapour carried over to the air pump, can be effected without any considerable allowance of cooling surface, but that if a further substantial reduction in air temperature is needed on account of the requirements of the air pump, then this would call for a substantial increase in cooling surface area. In other words, if the volumetric capacity of the air pump is insufficient to perform what may be called its legitimate duty, that is to say, to handle the air volume corresponding to the air temperature determined by the sole consideration of the maximum amount of vapour to be carried over, then any saving in first cost and running cost of the air pump may be dearly bought in the price of additional cooling surface in the condenser.

* *Zeitschrift des Vereins deutscher Ingenieure*, 5th July, 1919

Similarly, where test results indicate that a condenser is not performing the duty for which it is designed, this may be due to the fact that it is being called upon to cool the air to too great an extent, that is to say, that the fault lies with the air pump and not with the condenser.

The general design of a surface condenser is naturally governed by the requirements of its primary function of condensing steam, and not by those of cooling air.

The whole development of modern condenser designs is therefore based on the fundamental importance of securing as high a value as is possible for K_s , the heat transmission coefficient from the tubes to cooling water in conjunction with an efficient distribution of the steam in the steam space. A well-designed condenser cannot therefore be expected to act as an efficient air cooler, in which the importance of securing the highest rate of heat transmission from air

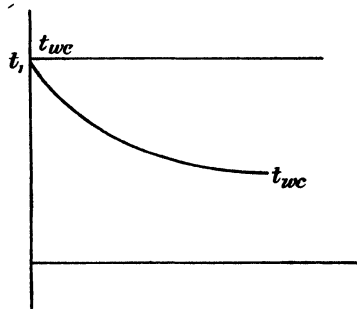


FIG 8
MEAN TEMPERATURE DIFFERENCE
FOR CONDENSATE.

to metal quite outweighs that of aiming at a high value for K_s . No doubt a full appreciation of this difference has led some condenser makers to interpose a separate piece of apparatus between the condenser and the air pump, and which can conveniently embody the essential features of design required to enable it efficiently to perform its duty as an air cooler.

Surface Area for Cooling the Condensate. The cooling of the condensate within the condenser is a practice frequently adopted when the condensate is withdrawn by the air pump, but the necessity for this does not arise where the condensate and air pumps are separate. In such cases the designer usually aims at obtaining as high a condensate temperature as possible. Where the condensate has to be cooled, this is effected by drowning the lower rows of tubes. Here again it is scarcely practicable to predetermine the amount of cooling surface required to produce a given cooling effect, but if in a given condenser the amount of cooling surface covered by the condensate is determined by a gauge glass reading and the temperature of the condensate before and after cooling, then we can say

$$Z_w = \frac{H_w}{d_{mw} K_w}$$

where Z_w = cooling surface required for cooling the condensate.

H_w = heat abstracted from the condensate = $Q_s (t_{we} - t_{wc})$

t_{we} = condensate temperature before cooling.

t_{wc} = condensate temperature after cooling.

$$d_{mw} = \frac{t_{we} + t_{wc}}{2} - t_1, \text{ or more accurately}$$

$$d_{mw} = \frac{D_w - d_w}{\log_e \frac{D_w}{d_w}}$$

where $t_{we} - t_1 = D_w$ and $t_{wc} - t_1 = d_w$

K_w = heat transmission coefficient from condensate to cooling water.

The circulating water temperature has been taken as constant as the rise in temperature is negligible, since it is equal to the drop in condensate temperature, divided by the ratio of the circulating water quantity to the steam quantity.

If K_w is determined from this formula, then we can calculate the cooling effect for other temperature conditions, and for the same surface with a reasonable degree of accuracy. But in using K_w for estimating results obtainable with a different design of condenser, allowance would have to be made for any variation in the shape of the condenser or of the spacing of the tubes. The resistance to heat transfer from the condensate to the tubes is very high in comparison to that from the tubes to the circulating water, on account of the low water velocity of the condensate and of the comparatively wide spacing of the tubes. The amount of cooling surface allowed for reducing the condensate temperature does not, however, usually exceed 5 per cent of the whole.

Division of Total Surface. It follows from the foregoing that the total surface may be divided into two or into three sections, namely Z , Z_a and occasionally Z_w , each of which serves its specific purpose. Since, however, Z_w is always small, and Z_a cannot in any case be predetermined, the designer will not gain any advantage by attempting to treat the three factors separately. On the other hand, it is advisable to subdivide the surface in this manner in comparing test results, providing that these indicate that Z_a or Z_w are of appreciable value in comparison with Z .

CHAPTER VII

SURFACE CONDENSERS—CALCULATION OF DIMENSIONS

THE surface area having been determined together with the water quantity and the water velocity, then the number of tubes and the tube length are fixed for a given diameter and gauge of tube, assuming a single water pass.

That is to say, if

Q_w = Water quantity in gals. per min

v = Water velocity in ft. per sec.

A_i = Internal tube area in sq. in.

then number of tubes in parallel

$$N = \frac{Q_w}{v \times A_i \times 60 \times 6.235} \times 144$$

$$= 0.385 \frac{Q_w}{v \times A_i}$$

The tube length $l = \frac{Z}{A_s \times N}$ ft.

where Z = surface in sq. ft.

and A_s = external surface area per linear foot in sq. ft.

The dimensions and quantities given in Table V will be found useful for determining N and l for the most usual diameters and thicknesses of tubes.

TABLE V

External tube diameter .	$\frac{1}{8}"$	$\frac{1}{4}"$	$\frac{3}{8}"$	$\frac{1}{2}"$	$1"$	$1\frac{1}{2}"$
S.W.G. .	18	19	18	19	18	19
Weight per linear foot, lbs	0.32	0.27	0.39	0.33	0.53	0.45
Weight per sq. foot, lbs. .	1.955	1.651	1.992	1.682	2.025	1.718
Thickness of tube .	0.048"	0.040"	0.048"	0.040"	0.048"	0.040"
Surface per linear foot, sq. ft. .	0.1635	0.1635	0.196	0.196	0.262	0.262
Internal diameter, in. .	0.529	0.545	0.654	0.67	0.904	0.920
Internal area, sq. in. .	0.22	0.233	0.336	0.3526	0.642	0.666
One foot per second = gallons per min. per tube	0.572	0.606	0.875	0.916	1.67	1.735

If the figures in the bottom line of Table V be called q , then the calculation of N can be simplified thus $N = \frac{Q_w}{qv}$.

Number of Water Passes. The tube length arrived at is generally altogether excessive for the condenser to be built with a single water pass. It is therefore necessary to subdivide the tube length into two or more passes. In modern practice, two or three passes are generally adopted for large condensers, and three or four passes for smaller sizes. A comparatively large number of passes emphasize the countercurrent principle which is, of course, entirely eliminated in a single pass condenser; on the other hand, a greater number of passes increase the circulating water head, and to some extent the first cost, in that it involves more expensive water ends and tube plates. The designer usually chooses the number of passes to be adopted on the basis of the most favourable ratio of diameter to length between tube plates, or of tube plate area to length if the tube plate is not circular. He will also be guided in his choice of number of passes by the effect on the lay out of the pipe lines. If the circulating water mains or ducts run parallel to the axis of the condenser, then he will usually find that the inlet and outlet pipes leading to and from the condenser can be arranged equally favourably for a condenser having an even number of passes and consequently having inlet and outlet branches on the same end as they can for one having an odd number of passes, that is to say, one whose inlet and outlet branches are placed on opposite ends. On the other hand, if, for instance, the water mains run at right angles to the condenser axis and both inlet and outlet mains are on the same side, then it is a matter of some importance to choose an even number of passes so as to avoid the necessity of bringing either the inlet or the outlet pipe right across the back of the condenser. Such an arrangement usually means encroaching on the limited space available for the turbine foundations and involves the introduction of one or more sharp pipe bends that are unsightly and represent a definite and practically continuous waste of power. The amount of power that is absorbed in a sharp pipe bend may be very minute when reckoned as a percentage of the power generated, or of the heat units handled by the whole set. All the same, the waste is an absolute one and should therefore be avoided if at all possible. The satisfactory arrangement of the water pipe lines, especially in the case of large condensing plants, is always a difficult problem and the general accessibility and neatness of appearance of the condenser pit depend very largely on the care bestowed upon this problem.

It is well to appreciate that the ease of supervision, and with that

the reliability of the plant, is greatly enhanced by ensuring that all condenser pipe work and valves and the auxiliaries are readily accessible and readily visible.

Ratio of Diameter to Length. The tube plate area or diameter in relation to the number of tubes varies with the spacings of the tubes and the amount of area allowed for steam lanes, and the marginal space of the tube plates opposite the water box division plates, and for stay bolts, etc. If the designer has laid out a number of tube plates for various sizes, based on his required spacings of tubes, he can construct suitable curves giving the plate diameter or area as a function of the number of tubes, and use these curves for the purpose of checking his ratio of tube length to plate area or diameter when the tube number has been calculated.

In choosing his ratio of condenser length to diameter the designer may be tied by considerations of available space, especially so in the matter of length, since, apart from the actual length of the condenser, allowance has to be made for withdrawing tubes so that every increase in actual tube length involves twice the corresponding addition to space occupied. A short condenser of large diameter or plate area will assist a good steam distribution, but again involves a greater capital cost in most cases on account of the larger tube plates and water boxes. As a general rule the ratio of length to diameter should be within the limits of 1.5 to 2.5.

EXAMPLE. To calculate the surface and dimensions of a condenser to deal with 10,000 lbs. of steam per hour and to maintain a vacuum of 28 ins. with cooling water at 70° F.

assuming $d = 9^\circ \text{ F.}$

$$\text{then } t_v = 101.11^\circ \text{ F.}$$

$$D = 31.11^\circ \text{ F.}$$

$$D-d = 22.11$$

If $H_1 = 1000 \text{ B.Th.U.'s per lb.}$

$$\text{then } Q_w = \frac{10,000 \times 1000}{22.11 \times 60 \times 10} = 757 \text{ gals. per minute, say,}$$

760 gals. per minute.

Let $v_w = 5 \text{ ft. per second}$

and $K = 400$

$$d_m = \frac{22.11}{\log_e \frac{31.11}{9}} = \frac{22.11}{\log_e 3.46} = \frac{22.11}{1.241} = 17.83$$

$$\text{then } Z = \frac{10,000 \times 1000}{400 \times 17.83} = 1400 \text{ sq. ft.}$$

For $\frac{3}{4}$ in. tubes, 18 S.W.G., $q = 0.875$

$$N = \frac{760}{5 \times 0.875} = 174$$

$$l = \frac{1400}{0.196 \times 174} = 41.2 \text{ ft.}$$

A four-flow condenser would be required to keep the length between plates within reasonable proportions, namely, $l_t = \frac{41.2}{4} = \text{say } 10 \text{ ft. } 3 \text{ ins.}$ The tube plate diameter of a circular condenser for these figures would probably be in the neighbourhood of 4 ft., so that the resultant ratio between length and diameter is rather high. If it is desired to build a shorter condenser with the same tube size without having recourse to five passes, then we can do so by reducing the water velocity or by increasing d . If $v = 4 \text{ ft.}$ then we should have to allow a smaller value for K , say $K = 380$.

Then $Z = 1475 \text{ sq. ft.}$

$$N = 218$$

$$l = 34.5$$

and length between plates = 8 ft. $7\frac{1}{2}$ ins.

If we retain $v = 5 \text{ ft. per second}$ and decide to increase d to 12° F.

then $D - d = 19.11^\circ \text{ F.}$

and $Q_w = 872 \text{ g.p.m.}$

$$d_m = 19.85$$

and $Z = 1260$

then $N = 200$

$$l = 32.2 \text{ ft}$$

EFFECT OF MODIFICATIONS. It will be seen that either modification gives the result aimed at, namely, a reduction in the value of l for a given duty and consequently a reduction in the ratio of length to diameter, frequently referred to as the surface section ratio. If v is decreased, then the size and cost of the condenser is increased slightly, but the auxiliary power consumption will be reduced.

If d is raised then the size and cost of the condenser will be reduced, but the water quantity required, and consequently the circulating pump power consumption, will be higher.

The greater cost of the larger pump, etc., needed, may easily nullify the saving due to the reduced surface area. The proposal to work with a lower water velocity would therefore appear to be more advantageous.

If the same conditions chosen for our example were made to apply to a condenser designed to deal with a larger steam quantity, say 20,000 lbs. per hour, then N will be proportionately increased to 348, but l will be unaffected. Under these conditions the original values for v and d can be retained without leading to an excessively high ratio of length to diameter in a four-pass arrangement, and in fact, a three-flow condenser would become practicable with a slight reduction in v or increase in the value of d .

CHAPTER VIII

SURFACE CONDENSERS—CALCULATION OF WATER HEAD

THE water head or loss of water pressure P_w across a condenser is made up of the static head P_{ws} and the friction head P_{wf}

$$P_w = P_{ws} + P_{wf}$$

In principle, the static head in feet of water is given by the vertical measurement in linear feet between the horizontal centre lines of the water inlet and outlet branches, providing that these represent the highest and lowest water points in the condenser. The effective value of P_{ws} as reflected in the head to be allowed on the circulating pump, depends on the arrangement of the circulating water system, and we have to consider the following series of possible conditions.

First Case. Water drawn from a river or canal, where the suction inlet level is at or above the level of the discharge, and the condenser outlet branch is not above the point of discharge. In this case there is no loss of head due to the height of the condenser or the vertical distance between inlet and outlet branches, since this head is compensated by a corresponding negative head on the suction side of the pump, and the total head on the pump is that due to friction.

Second Case. Water supply level conditions as in case (1), but the condenser outlet or both inlet and outlet are above the supply level. If the outlet pipe can be sealed, and a syphon can be maintained, then again no static head needs to be allowed for. If, on the other hand, no syphon is maintained in the circulating water system, then the vertical distance between the water level and the centre line of the water outlet gives the value of P_{ws} to be allowed in the calculation of the water head.

Third Case. If the water supply is pumped from a low level through the condenser to a point at a level above the condenser outlet, a condition which usually obtains in cooling tower installations.

In this case the height of the condenser is again immaterial to the calculation of the total head on the pump.

The conditions under which the height of the condenser affects the water head on the pump arise therefore only under Case (2). The theoretical maximum syphonic head that can be maintained is equal to that of the atmospheric pressure prevailing. In practice the maximum is hardly obtainable.

Loss of Head Due to Friction. The friction head may be divided into three headings, namely: First, that due to the water ways, that is to say the inlet and outlet branches and the water boxes; second, the ferrule and tube orifice friction, and third, the friction within the tubes. The last-named can be predetermined with some degree of accuracy.

CALCULATION OF FRICTIONAL RESISTANCE IN TUBES. Messrs. Stanton & Pannell* have carried out a series of experiments at the National Physical Laboratory, from which they found that the frictional resistance of any fluid passing through a clean pipe is given by the formula $F = \rho v^2 L$, where F = resistance in dynes per sq. cm. of internal area (1×10^6 dynes per sq. cm. = 14.5 lbs. per sq. in.) and L varies with the nature of the fluid, and according to Professor C. H. Lees,† L can be determined with accuracy by the expression

$$L = 0.0763 \left(\frac{\mu}{\rho v d} \right)^{0.35} + 0.0009$$

where μ = viscosity of the fluid in C.G.S. units, i.e. dynes per sq. cm.

ρ = the density of the fluid in gr. per cu. cm.

v = velocity in cm. per sec.

d = internal diameter of pipe in cm.

Table VI gives the values for F in feet per 100 ft. tube lengths as calculated from these formulae for $\frac{1}{8}$ in., $\frac{3}{4}$ in. and 1 in. tubes of No. 18 S.W.G., and for mean water temperature 50° F., 68° F., 86° F., 104° F. and 122° F., and water velocities of 3, 4, 5, 6 and 7 ft. For convenience, F is here expressed in feet of water. The corresponding curves for $\frac{3}{4}$ in. tubes have been plotted in Fig. 9, and intermediate values can be read from these curves. (See Table VI, page 53.)

As the figures in Table VI are based on distilled water and clean tubes, it is advisable to allow a margin of 15 to 30 per cent in using these values, according to the nature of the water.

It should be noted that when scale is deposited in the tubes their effective diameter decreases so that an additional correction has to be introduced.

The formulae used are somewhat cumbersome, and they can be simplified for general practice without seriously impairing their usefulness. As far as fresh water is concerned ρ can be taken as unity

* *Phil. Trans., A.*, Vol. CCXIV, 1913-14.

† *Proc. Roy. Soc.*, Vol. XCI.

throughout, and instead of using the 0.35th power in the expression for L , we can take the cubic root. F can then be determined by

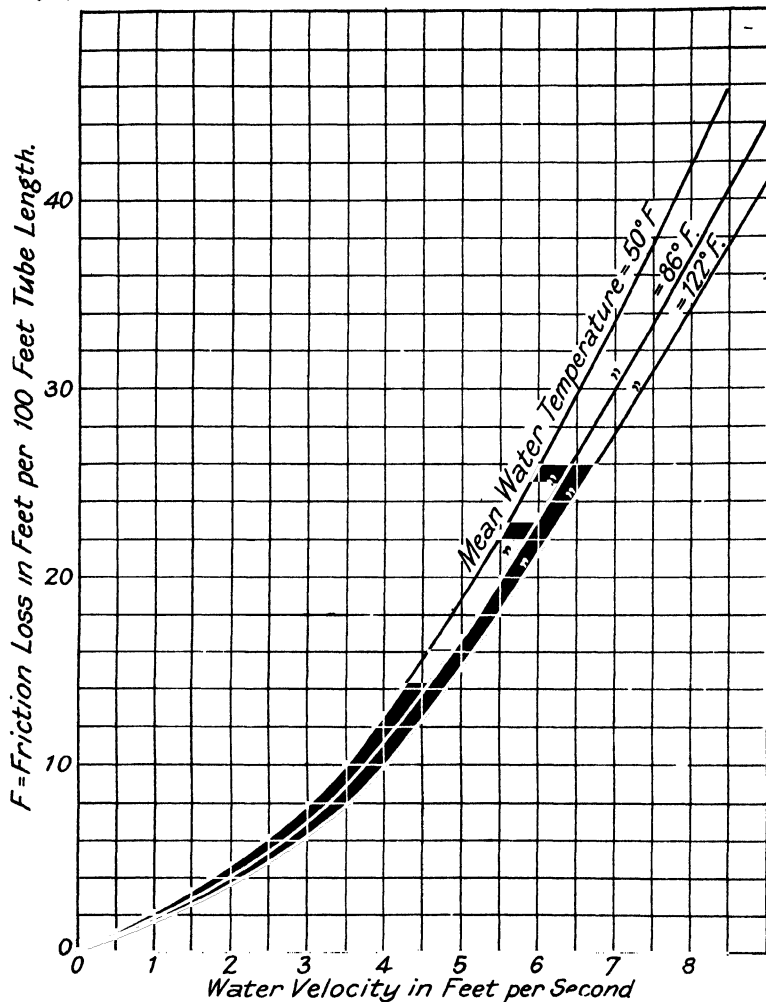


FIG. 9.

FRICTION LOSS ON 100 FT. LENGTH OF $\frac{3}{4}$ IN. TUBE.

slide rule calculation without the use of logarithms. The formulae would then read

$$F = v^2 L$$

$$L = 0.0763 \sqrt{\frac{\mu}{\rho v d}} + 0.0009$$

EXAMPLES SHOWING NATURE OF ERROR DUE TO USE OF SIMPLIFIED FORMULAE. If we take two fairly extreme examples and compare the result obtained, we can judge the magnitude of the error involved.

We shall first assume distilled water at a mean temperature of 110° F. passing through 1 in. No. 18 S.W.G. tubes at a velocity of 3 ft. per sec.,

$$\text{then } d = 0.904 \text{ in.} = 2.3 \text{ cm.}$$

$$\mu = 0.00617$$

$$\rho = 0.991$$

$$v = 91.4 \text{ cm. per sec.}$$

$$\text{from } L = 0.0763 \left(\frac{\mu}{\rho v d} \right)^{0.35} + 0.0009$$

$$L = 0.0763 \left(\frac{0.00617}{0.991 \times 91.4 \times 2.3} \right)^{0.35} + 0.0009$$

$$L = 0.002884$$

$$F = 0.991 \times 91.4^2 \times 0.002884$$

$$F = 23.95 \text{ dynes per sq. cm. wetted surface}$$

$$\text{or } F = 4.175 \text{ ft. per 100 ft. tube length}$$

$$\text{from } L = 0.0763 \sqrt[3]{\frac{\mu}{v d}} + 0.0009$$

$$L = 0.00326$$

$$F = 91.4^2 \times 0.00326 = 27.2$$

$$\text{or } F = 4.74 \text{ ft. per 100 ft. of tube}$$

$$\therefore \text{error} = \frac{4.74}{4.175} = 1.14 \text{ or } 14 \text{ per cent.}$$

Next, if we assume distilled water at a mean temperature of 50° F. passing through $\frac{5}{8}$ in. tubes at a velocity of 7 ft. per sec.,

$$\text{then } d = 0.529 \text{ in.} = 1.344 \text{ cm.}$$

$$\mu = 0.01311$$

$$\rho = 0.9997$$

$$v = 212.7 \text{ cm. per sec.}$$

then, using the correct formula,

$$L = 0.003211$$

$$F = 44.63 \text{ ft. per 100 ft.}$$

using the approximate formula

$$F = 0.00363$$

$$F = 49.6 \text{ ft. per 100 ft. of tube}$$

$$\therefore \text{error} = \frac{49.6}{44.03} = 1.13 \text{ or } 13 \text{ per cent.}$$

FRICTION LOSS IN BOXES AND TUBE ENDS. Unfortunately, no general rules or formulae can be evolved for the determination of

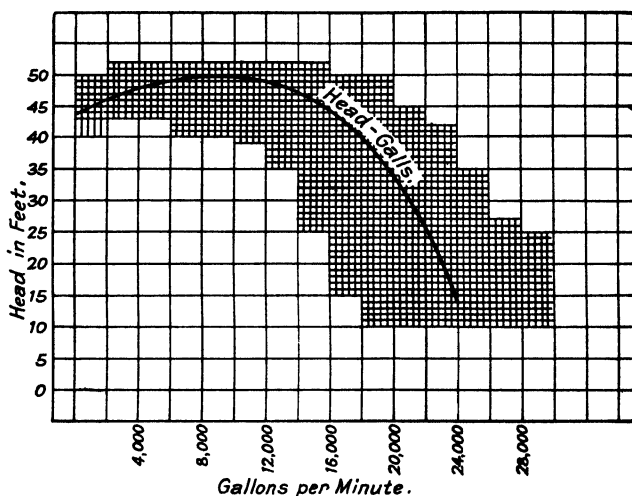


FIG. 10.

CENTRIFUGAL PUMP CHARACTERISTICS—NORMAL DELIVERY
20,000 GALS. PER MIN. AT 30 FT. HEAD.

the friction loss in water boxes, tube ends, etc., as the friction head depends largely on the shapes adopted. For a given design the friction loss will, no doubt, again be a function of the water velocity and temperature. By avoiding abrupt changes of direction of flow and restrictions of the water passages, the designer can keep these losses down to reasonable proportions. In the absence of reliable test results, the designer can arrive at a fair approximation by assuming the external losses per pass as equal to the loss within the tubes. Since the internal loss varies with the length of tube, and the external loss is entirely independent of this factor, the result can obviously only be taken as a very rough approximation.

Circulating Water-pump Characteristics. Where the water head is variable as, for instance, in the case where the water is drawn from

a tidal river, where the maximum head at low tide exceeds the syphonic maximum, the problem of maintaining a constant water supply presents great difficulties, as the maximum water quantity delivered by a centrifugal pump varies with the head.

Where a syphonic system is in use, the pump characteristic should be such that the pump is still capable of delivering a reduced quantity of water against the increased head.

Fig. 10 shows the characteristics of a pump designed normally to deliver 20,000 gals. of water per minute against a total head of 30 ft. If, for example, this head is increased to 45 ft. owing to the failure of the syphon, then the water quantity will fall to 15,000 gals. per minute, entailing a substantial fall in vacuum but not a total failure. If, however, the breakdown of the syphon meant raising the total head to 55 ft., then the pump would no longer be capable of delivering any water. In other words, a pump with this characteristic should not be used where the assumed conditions prevail. Variations in head can, of course, be dealt with to some extent by throttling and by speed variations, but neither method lends itself readily to regular application in practice.

TABLE VI
FRICTION LOSS PER 100 FT. OF TUBE No. 18 S.W.G.

Water Velocity through Tubes, ft. per sec.	3 ft	4 ft	5 ft	6 ft.	7 ft.
$\frac{5}{8}$ in. Tubes—					
50° F.	10·10	16·61	24·49	33·63	44·63
68° F.	9·44	15·56	22·96	31·57	41·39
86° F.	8·89	14·68	21·69	29·85	39·16
104° F.	8·46	13·98	20·68	28·50	37·41
122° F.	8·09	13·40	19·84	27·67	35·95
$\frac{3}{4}$ in. Tubes—					
50° F.	7·70	12·69	18·73	25·74	33·74
68° F.	7·21	11·90	17·59	24·20	31·75
86° F.	6·79	11·24	16·63	22·91	30·09
104° F.	6·47	10·72	15·87	21·90	28·78
122° F.	6·20	10·10	15·25	21·04	27·68
1 in. Tubes—					
50° F.	5·05	8·35	12·33	16·98	22·28
68° F.	4·74	7·84	11·60	16·00	21·10
86° F.	4·47	7·42	10·99	15·18	19·95
104° F.	4·27	7·09	10·51	14·52	19·11
122° F.	4·09	6·81	10·11	13·98	18·41

CHAPTER IX

SURFACE CONDENSERS—CONSTRUCTIONAL DETAILS

Condenser Tubes. Condenser tubes are generally made of various grades of good quality brass or of other copper alloys. The most commonly used mixture consists of 70 per cent copper, 30 per cent zinc. For brackish or salt water, so called Admiralty mixture of 70 per cent copper, 29 per cent zinc and 1 per cent tin has been found to give better results, but is somewhat more costly. Many users prefer to tin the standard 70:30 tubes. This practice is frequently effective in preventing corrosion so long as the tinning remains undisturbed, but it naturally loses its efficacy as soon as the tin is worn away by erosive effects of the water itself or of foreign bodies carried along with the water.

MANUFACTURE OF TUBES. The usual method of manufacture is to cast the metal in Plumbago crucibles using, say, 25 per cent of its own scrap, and pouring into cast iron billets with a core in the centre, but special care is required to ensure good sound castings. The core is made of wood fibre or hay with loam admixture, and must be perfectly dried in drying stoves. The core bar, which is hollow, must be perforated to allow of the escape of gases.

Special care has to be exercised in drying the core, as the use of damp cores has proved a very frequent source of trouble in the production of tubes.

The dimensions and lengths of castings vary with works practice and are, say, from 3 ft. to 4 ft. 6 ins. long, $2\frac{1}{2}$ ins. outside diameter, and about $\frac{1}{2}$ in. thick.

The billets are occasionally required to be turned on the outside and bored inside. The advantages, however, of bored and turned billets are questionable as, apart from being costly, the operation removes the natural skin from the inside of the tube. Some authorities maintain that this skin helps to resist corrosion.

The billets are cold-drawn over mandrels and through dies on drawbenches. At each stage the tubes must be annealed and great care has to be exercised, as on this operation largely depends the production of satisfactory tubes.

TESTING TUBES DURING MANUFACTURE. It is usual to work to a specification of plus or minus $2\frac{1}{2}$ per cent above or below the theoretical weights of tubes when measured on tube lengths of

100 ft. The test pressure varies, but an internal hydraulic pressure of 600 lbs. per sq. in. is not generally considered excessive. For the purpose of inspection it is usual to group the tubes into lots of 100, and to take one tube from each parcel for the following test.

Each of the selected tubes is carefully examined, then cut in halves, and the ends of the tubes thus formed must stand being raised to red heat without flying or splitting; when cold, these ends must stand flattening close without cracking.

CALCULATING BURSTING PRESSURE OF TUBES. To consider the mechanical strength of a tube we have to calculate the bursting stress, due to the sum of the water head inside and the vacuum outside the tube. For this purpose we can use the usual formula

$$\frac{2T \times S}{D} = H$$

Where H = head in lbs. per sq. in.

T = thickness of tube in ins.

D = inside diameter of tube in ins.

S = tensile load on the material in lbs. per sq. in.

We shall take what amounts to an extreme case where it is proposed to use 1 in. tubes, No. 19 S.W.G., and a total water head of 30 lbs. gauge per sq. in.—a figure which is only likely to occur where the water is pumped through the condenser to the inlet of a cooling tower, and is passed through a considerable length of piping between the condenser and the tower. We shall further assume a vacuum of 14 lbs. per sq. in.

We then have

$$T = 0.040 \text{ in.}$$

$$D = 0.920 \text{ in.}$$

$$H = 44 \text{ and } S = \frac{44 \times 0.92}{2 \times 0.040} = 506 \text{ lbs. per sq. in.}$$

This value gives a sufficient factor of safety for the materials employed, and since the conditions here assumed represent an extreme case, we can conclude that there is no need to calculate the tube bursting stress in individual condenser designs.

DETERMINING DEFLECTION OF TUBES. To calculate the deflection

of a condenser tube we can consider this as an evenly loaded beam supported at both ends, for which the maximum deflection

$$\Delta = \frac{5}{384} \frac{Wl^3}{EI} \text{ where}$$

Δ = maximum deflection in ins.

W = total load in lbs. (in this case weight of tube plus weight of water in tube).

l = unsupported length of tube in ins.

E = modulus of elasticity, say, 10×10^6 for brass tubes.

I = moment of inertia for a tube $\frac{\pi}{64} (D^4 - d^4)$

D = outside diameter of tube in ins.

d = inside diameter of tube in ins.

If we take the case of a $\frac{3}{4}$ in. tube, No. 19 S.W.G., 10 ft. long, then
Weight of water = $0.3526 \times 10 \times 12 \times 0.036 = 1.54$ lbs.

$$\text{Weight of tube} = \frac{\pi}{4} (0.75^2 - 0.67^2) \times 10 \times 12 \times 0.3 = 3.3 \text{ lbs.}$$

$$W = 4.85 \text{ lbs.}$$

$$I = \frac{\pi}{64} (0.75^4 - 0.67^4) = 0.00565$$

$$\Delta = \frac{5}{384} \times \frac{4.85 \times 120^3}{10 \times 10^6 \times 0.00565} = 1.97 \text{ ins.}$$

A deflection of this magnitude is excessive and, in point of fact, an unsupported tube length of 5 ft. is usually considered as a maximum. The corresponding deflection at the centre for the same tube 5 ft. long would be 0.123, or roughly $\frac{1}{8}$ in.

EXPANSION OF TUBES IN CONDENSER. If we take the linear coefficient of expansion of brass at 1.04×10^{-5} per degree F., then the maximum expansion per foot run of tube is that due to a temperature of 212° F., which may occur in the event of a failure of the vacuum. Assuming a minimum temperature of 32° F., then the total expansion will be $1.04 \times 10^{-5} \times 180 \times 12 = 0.0225$ in. per linear foot of tube. The normal temperature variation is, of course, very much lower, and the normal expansion correspondingly less.

VIBRATION OF TUBES IN CONDENSER. Trouble occasionally arises through the tubes vibrating in synchronism with some regular

pulsation about the plant. The natural frequency of vibration of the tubes may be calculated as follows.

In the calculation of the deflection it is obvious that the deflection is proportional to the load, and it may be said that with a load of n lbs. the deflection is 1 in. When deflected 1 in., the tube is opposing further deflection by a force of n lbs. and certain energy is stored in the tube. There is also some kinetic energy in the tube and the sum of these two items remains constant, or it may be so assumed, for calculation purposes, thus

$$E = \frac{1}{2}mv^2 + \frac{1}{2}nx^2$$

where m = moving mass

x = deflection

$$v = \text{velocity of tube} = \frac{dx}{dt}$$

Differentiating and simplifying, it becomes

$$\frac{d^2x}{dt^2} + \frac{n}{m}x = 0$$

which is the usual equation for harmonic motion. The solution is

$$x = A \sin (qt + a)$$

where A and a are constant and q is equal to $\sqrt{\frac{n}{m}}$. But q also is equal to f the frequency of vibration $\times 2\pi$, so

$$f = \frac{1}{2\pi} \sqrt{\frac{n}{m}}$$

In the above case

$$n = \frac{4.75}{1.93} = 2.46$$

$$m = 4.75/32.2$$

$$\text{hence } f = \frac{1}{6.28} \sqrt{\frac{2.46 \times 32.2}{4.75}} = 0.65 \text{ vibrations per second}$$

= say, 40 vibrations per minute.

It is important to remember that not only would this tube be set into resonant vibration if it were subjected to an impulse of this frequency, but that overtones of this frequency would also have the same effect.

Halving the free length of the tube reduces the deflection in the ratio of 2^4 to 1 or to 16 to 1. Hence, for the unit deflection of 1 in. the value of n is increased 16 times. At the same time the value

of m is halved so that f is increased in the ratio of $\sqrt{\frac{16}{0.5}}$ or 5.65 to 1.0, giving a natural frequency of vibration for 5 ft. tubes of 226 per minute.

Tube Plates. The tube plates are generally made of rolled brass or of Muntz metal in this country and in the States. Continental makers use steel plates very largely for condensers using fresh water, though for marine purposes the use of steel for this purpose does not appear to have been adopted by any maker. Wherever practicable, the tube plates are made in a single piece to avoid the necessity of introducing joints.

The thickness of the tube plates varies from a minimum of $\frac{5}{8}$ in. to a maximum of $1\frac{3}{8}$ ins. The usual pitching of the tubes for $\frac{3}{4}$ in. diameter varies from $1\frac{1}{16}$ in. to $1\frac{3}{16}$ in. or $1\frac{1}{4}$ in. Close pitching has the very obvious advantage of reducing the diameter of the plates and of the shell for a given surface and length, and therefore represents a substantial saving in cost and in space efficiency. Unless, however, ample provision is made for effective steam lanes which affect the design in the opposite direction, too close pitching impedes the flow of the steam and of the air. Cases have come to the authors' notice where the removal of rows of tubes for the purpose of providing steam lanes had the effect of increasing the overall heat transmission coefficient of the condenser to the extent of improving the vacuum registered, notwithstanding the reduction in the total surface area.

Designers frequently adopt a closer tube pitching in the lower portion of the condenser where the steam volume is reduced, than in the upper section where the full steam volume has to be handled.

The load on the tube plates due to water pressure is equal to that on the condenser doors plus that due to the vacuum prevailing on the steam side of the plates, and, as in the case of the doors, we can assume the load to be evenly distributed over the whole area.

THICKNESS OF TUBE PLATE REQUIRED. To arrive at the thickness of tube plate and the number of stays required, we have to arrive at an equivalent section of solid plate in place of the section of plate between the glands or stuffing boxes, and we can then base the strength of the plate on the unsupported area, either between stays where these are employed, or between the supporting flanges and ribs. As already mentioned, the load is assumed to be evenly distributed over the unsupported area, and for rolled brass it is advisable to reduce the stress to a maximum of 7,000 to 8,000 lbs. per sq. in.

The following gives some typical figures for the equivalent solid

thickness for various plate thicknesses and tube pitchings for $\frac{3}{4}$ in. outside diameter tube—

Tube Plate Thickness.	Pitch	Equivalent Solid Plate Thickness.
1 in.	$1\frac{1}{8}$ in.	.482 in
$1\frac{1}{4}$ in.	$1\frac{1}{8}$ in.	.561 in
$1\frac{1}{2}$ in.	$1\frac{1}{8}$ in.	.543 in
$1\frac{3}{8}$ in.	$1\frac{1}{8}$ in.	.635 in
$1\frac{1}{2}$ in.	$1\frac{1}{8}$ in.	.704 in

In addition to checking the strength of the plates in the manner described, it is as well to bear in mind that the plate must be sufficiently rigid in its unsupported condition to be handled for machining and erection purposes, without deflecting beyond its elastic limit.

TYPICAL FORMS OF CONSTRUCTION. Fig. 11 shows a common method of securing the tube plates between the shell and the waterways. The jointing material consists generally of red lead and tape. Rubber packing is frequently adopted by continental makers. The figure also shows a method of securing the stay rods.

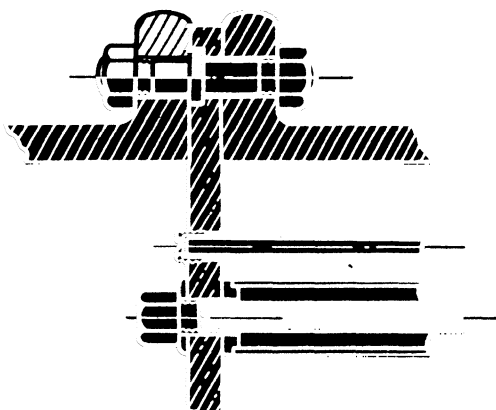


FIG 11
USUAL METHOD OF SUPPORTING TUBE PLATES
BETWEEN SHELL AND WATERWAY

Fig. 12, due to Messrs. Worthington-Simpson, shows the details of the connection between tube plate and stay rod for the 20,000 sq. ft. condenser described later, together with the method adopted for passing the rods through the tube support plates. The rods are made of Low Moor iron.

Fig. 13 shows a design of tubular longitudinal plate supports employed by Messrs. Escher Wyss, together with the method of securing the supports to the tube plates.

Fig. 14 shows the layout of a tube plate of a rectangular 7,000 sq. ft. condenser built by Messrs. Worthington-Simpson.

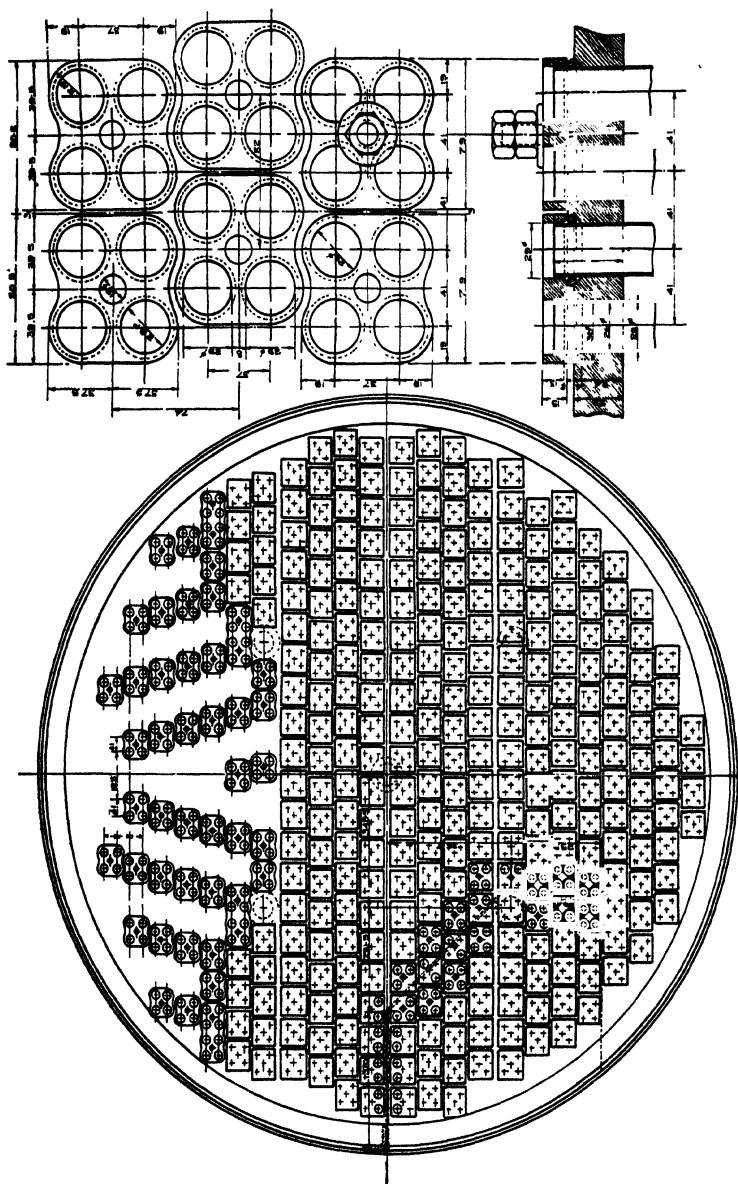


Fig. 15 (see p. 60).
TUBE PLATE ARRANGEMENT OF CONDENSER BY MESSRS. DORTMUNDER VULCAN.

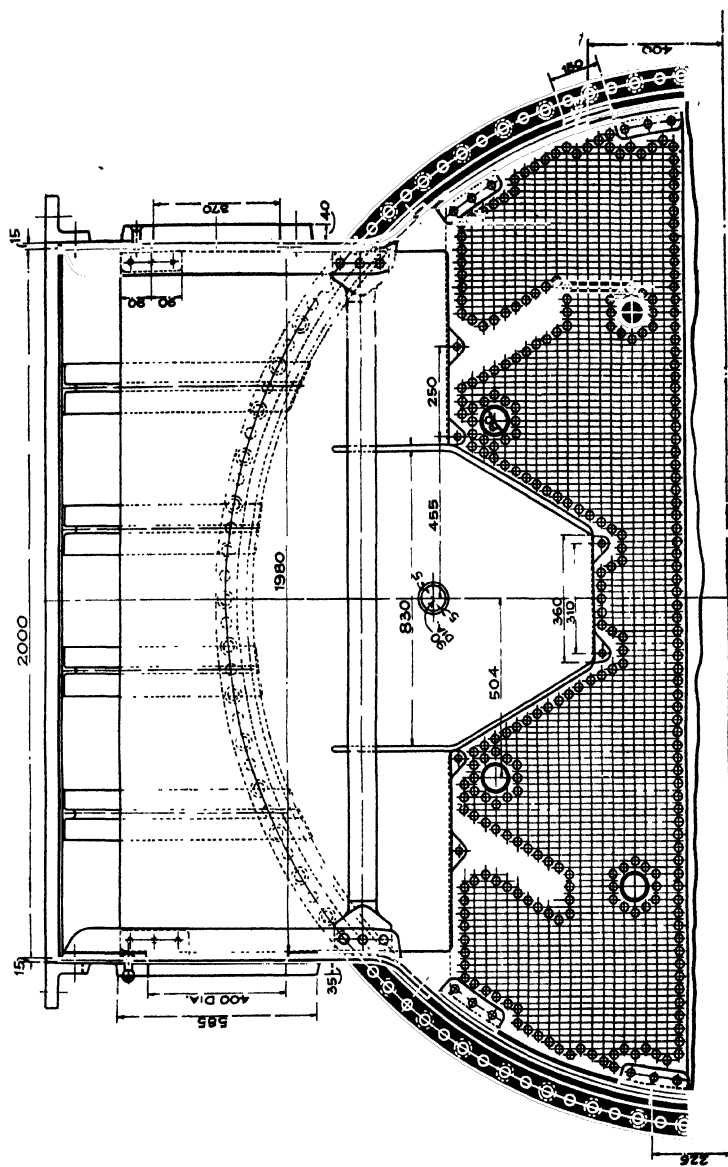


FIG. 16 (see p. 65).
ESCHER WYSS TUBE PLATE.

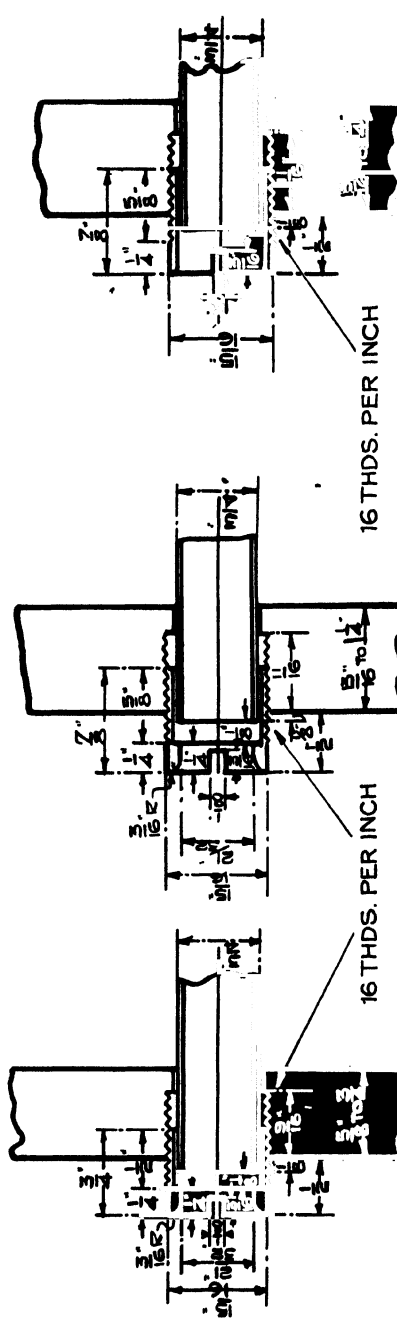


FIG. 17

RICHARDSONS-WESTGARTH TUBE GLANDS

In the case of Messrs. Escher Wyss's design shown in Fig. 16, the steam distribution is improved by the use of baffle plates in the centre with steam lanes at either side.

Tube Glands. The tubes require to be held in tube plates in such a manner as to allow them to expand and contract with variations in temperature without leakage of water from the water space into the steam space. As the glands used for this purpose form the points of entry and outlet for the tubes, it is advisable to make them bell-mouthed to reduce the friction loss.

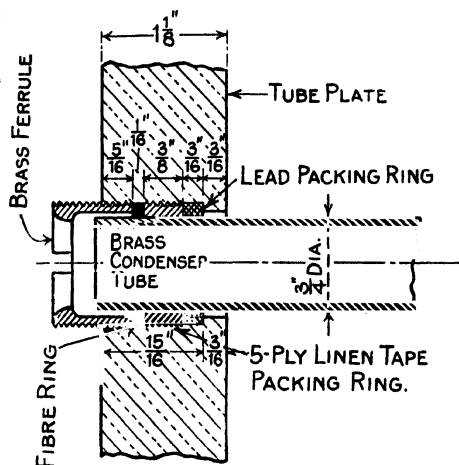


FIG. 18.

WORTHINGTON-SIMPSON TUBE GLAND
WITH ELECTRICAL CONTACT WASHER

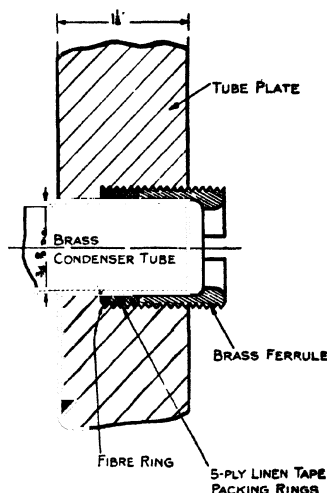


FIG. 19 (see p. 66).

WORTHINGTON-SIMPSON TUBE GLAND.

The friction loss at the tube inlets and outlets with the usual form of gland must represent a waste of power which would seem well worth attention. Isolated attempts to reduce this have not met with much success, owing to the fact that they involved the use of wider tube pitching, with the result that the additional cost outweighed the advantage gained. The space available with normal pitching is probably too cramped to give scope for anything but the simplest forms of gland.

The designs shown in Fig. 17, due to Messrs. Richardsons-Westgarth, of Hartlepool, are typical of the type of glands most usually adopted in this country and in America.

The packing is formed of grummets consisting of sized linen or wick which are compressed by brass ferrules screwed into the tube

plates. The ferrules are generally provided with a lip which limits the axial movement of the tubes in either direction. Messrs. Worthington-Simpson introduce a fibre ring at the back of the grummets as shown in Fig. 19. Where it is intended to ensure electrical contact between the tubes and the plates for the application of electrolytic protection devices, it is usual to fit lead washers at the back of the grummets. (See Fig. 18.)

Messrs. The Alberger Pump & Condenser Company of New York expand one end of the tubes into the tube plate. In the other end the tubes are packed with fibre which is held in position by ferrules

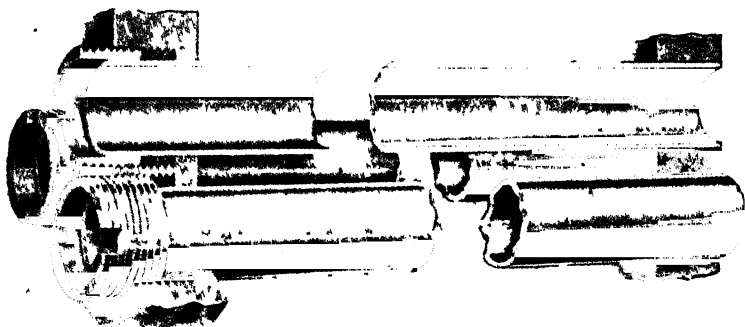


FIG. 20.
TUBE GLANDS.

(Alberger Pump and Condenser Co.)

without lips. The whole of the movement due to expansion and contraction must, of course, take place at this end. This method is illustrated in Fig. 20.

Screwed ferrules of the type described cannot be used to advantage where steel tube plates are employed, as the development of rust would damage the threads and prevent the effective unscrewing of the ferrules and would lead to leaky joints.

The design shown in Fig. 21 is used by Messrs. Escher Wyss, of Zurich, as an alternative to ferrule packing. It will be seen that the tubes are expanded at one end and slide in a rubber ring packing at the other end.

Rubber rings also form the packing medium in the design shown in Fig. 22, due to Messrs. Dortmunder Vulcan, of Dortmund, Germany. In this case the rubber rings are held in position by means of cast iron clamping boxes. Each box is arranged to

clamp four or more tubes. The same design is used by the Maschinenfabrik Gervembroich, of Gervembroich, Germany.

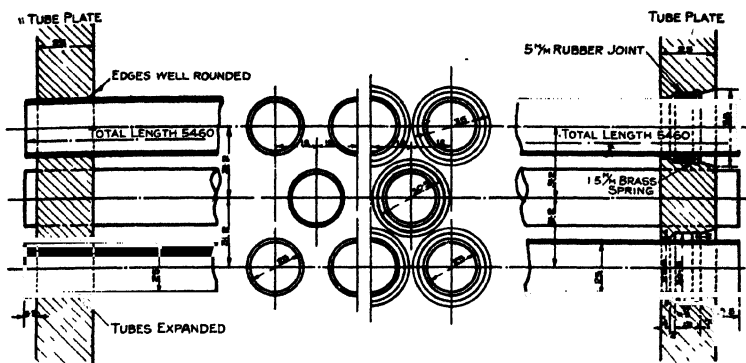


FIG. 21.
ESCHER WYSS RUBBER RING-PACKED GLANDS.

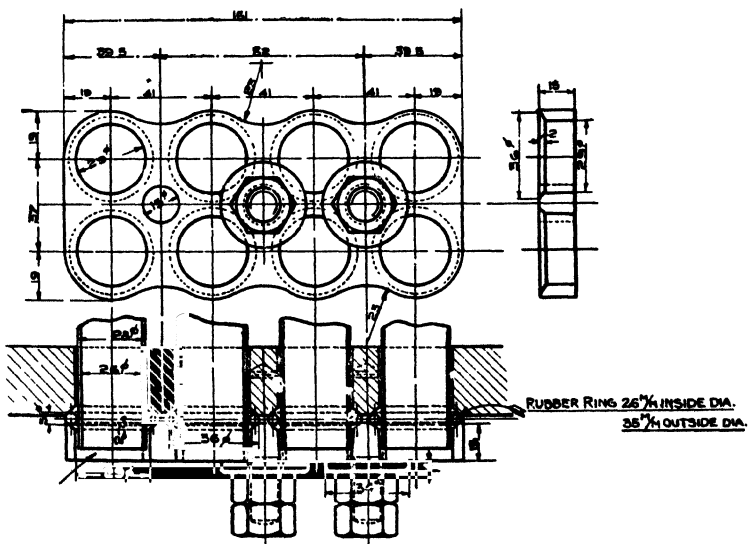


FIG. 22.
TUBE GLANDS USED BY MESSRS. DORTMUNDER VULCAN.

Shells. Condenser shells are made of cast iron or of steel. Cast iron is more frequently used in this country and in the United

States for land work, though marine practice and general continental practice favours the use of mild steel. For land work the question of price is no doubt the deciding factor in the choice of material in most cases, though for marine work the advantage of steel in the matter of weight is a matter of consideration. There are certain technical advantages in favour of either material. Thus, small and medium size cast iron shells can be formed in one piece, thereby avoiding joints within the shell. Steel shells are subject to corrosion, but are free from the danger of cracking when subjected to sudden variations of temperature, such as occur when changing from atmospheric to vacuum conditions and vice versa.

STRESSES ACTING ON SHELL. The normal stressing of the shell is that due to the load of the external atmospheric pressure against the vacuum within the condenser. When the prime mover is exhausting to atmosphere, the condenser may be placed under internal pressure due to the weight of the column of steam in the exhaust pipe line, but this loading is never likely to exceed 5 to 10 lbs. per sq. in.

It is obviously impracticable to test a condenser shell for external loading more than by a fractional amount beyond the normal loading. For the purpose of a test, the vacuum within the condenser can usually be raised slightly above that under which the condenser is likely to work, but the application of an external pressure above the atmospheric would involve exorbitantly costly and cumbersome plant. For this reason it is customary to rely on an internal pressure test of 30 lbs. per sq. in., and the designer has to allow for this test in calculating the strength of the shell as well as that of the tube plates.

REQUIRED THICKNESS OF SHELL. If a circular shell is considered as a simple tube then the bursting pressure is given by $p - \frac{2ft}{d}$, (1) where p is the pressure in lbs. per sq. in., f the safe working stress of the material in lbs. per sq. in., t the thickness and d the diameter in ins.

If we take f for cast iron at 3,000, then the dotted line in Fig. 23 gives the thickness of the shell derived from this formula and based on $p = 30$. These values for t are useless for practical purposes for two reasons: firstly, they are impracticable for even the best foundry practice, and, secondly, the resultant shells would be too weak to be handled in transport.

The designer must therefore use empirical figures with due consideration for the facilities and limitations of the foundry that will produce the casting. The fully drawn line in Fig. 23 gives

average values for t as a function of d derived from a large series of shells that have come under the authors' notice. Where the exhaust opening is large, vertical ribbings extending from the exhaust flange are usually introduced to assist the structure to support the proportion of the weight of the turbine casing that it has to bear.

The thickness of plates employed in steel condenser shells varies from about $\frac{3}{8}$ in. to $\frac{5}{8}$ in. Very thin plate shells are employed for

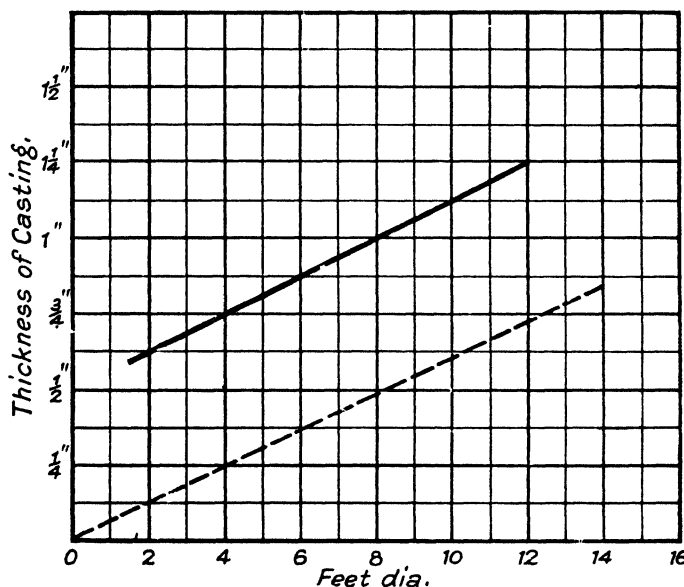


FIG. 23

THICKNESS OF CASTINGS FOR CYLINDRICAL CONDENSER SHELLS.

Admiralty work where reduction of weight is a matter of primary consideration. In using formula (1) for calculating the bursting strength of a steel shell, we can take $f = 10,000$ lbs. per sq. in., and allow 70 per cent efficiency for the riveted joints. An allowance of, say, $\frac{1}{16}$ in. should be made to compensate for weakening of the plates due to corrosion.

Where steel condensers are not of cylindrical shape, and flat surfaces are exposed to the test pressure, it is usual to reinforce the plates with T irons round the girth of the shell. In such a case we can neglect the loading on the plates, and consider the T irons

as uniformly loaded beams supported at the ends, the load being calculated as

$$W = P \times l \times a$$

where W = total load in lbs.

P = test pressure in lbs. per sq. in.

l = unsupported length of T iron in ins.

a = distance between centres of T irons in ins.,
and the greatest bending moment occurs at the middle of the beam
= $\frac{1}{8} Wl$.

Where it is intended to reduce the plate thickness to a minimum it is advisable to check the design for collapsing strength. For this purpose we can use Unwin's formula for strength of boiler flues, with longitudinal or cross joints—

$$P = 3.1094 \times \frac{10^6 t^{2.35}}{l^9 d^{1.16}} \quad (2)$$

Where P is safe working load in lbs. per sq. in., based on a factor of safety of 5

t = thickness of plates.

l = length between strengthening rings in ins. in our
case = length between tube plates.

d = diameter in ins.

We will take the case of a shell 8 ft. diameter, 12 ft. between tube plates and designed to stand a test pressure of 30 lbs. per sq. in. Then if we calculate t for bursting strength on the basis of (1), then

$$30 = \frac{10,000 \times t \times 70}{96 \times 100}$$

$$t = \frac{2880}{37,000} = 0.4114 \text{ in.}$$

Calculating on the basis of collapsing strength, and taking the external load as 14.7 lbs. per sq. in., we have from (2)

$$t = \sqrt[2.35]{\frac{14.7 \times 144 \times 96^{1.16}}{3.1094 \times 10^6}} = 0.335 \text{ in.}$$

It will be seen that the internal stressing is higher in the case of our present example, and demands a greater thickness of plate than is required if basing the thickness on collapsing strength.

Section of Exhaust Flange. A circular shape is usually adopted for the exhaust steam opening of small condensers, especially where

a long pipe line is required, as no change in pipe section is then required. Where a sluice valve is placed immediately over the condenser inlet it is naturally easier to deal with a circular opening. On the other hand, in the case of large high vacuum condensers, a circular opening does not give a satisfactory initial steam distribution, and calls for a dome-shaped steam space at the top of the condenser to give the necessary distribution of the incoming steam over the full length of the condenser. This means additional head-room in the condenser pit, and we therefore find that most large condensers have a rectangular opening extending over practically the whole length, and this in turn is usually arranged to conform to the exhaust flange dimensions of the turbine so that the two flanges can be bolted together without any intermediate piping. In the absence of an expansion pipe, the condenser body is then mounted on springs which take up the expansion due to variations in temperature. The size of the exhaust opening is determined by the permissible steam velocity. Modern turbine practice favours velocities of 300 to 400 ft. per second, the higher value applying to very high vacua. The question of the determination of the size of exhaust pipe lines in central condensers serving a number of engines is very fully discussed by Weiss in his work on condensers.*

Water Heads and Doors. Where the number of water passes is even, and where consequently the water outlet and inlet branches are at the same end of the condenser, designers frequently use a single waterway with inlet and outlet branches, and form the necessary guide passage from pass to pass in the door at the opposite end. In exceptional cases the waterways are omitted entirely, and the inlet and outlet branches are formed within the condenser doors. This arrangement has the disadvantage of necessitating breaking pipe joints when the doors are removed for cleaning the tubes. Cast iron waterways and doors are frequently adopted even where steel shells are used. The shaping of the inlet and outlet branches is a matter of great importance in keeping down the water friction. The doors are usually provided with numerous manhole or inspection doors. It is a great convenience to the user to have these inspection doors hinged and not bolted, as this minimizes the time and labour of opening up, and this method is very generally adopted in modern designs as shown in the photograph of a condenser built by Messrs. Allis Chalmers Manufacturing Co., of Milwaukee, Fig. 24. The constructional details of the hinged inspection doors of a Worthington-Simpson condenser are illustrated in Fig. 25. Where the main doors are made of steel plates it is customary

* F. J. Weiss, *Die Kondensation* (J. Springer, 1901).

to strengthen them with T irons or by carrying the tube plate stay-rods through to the doors

TOTAL LOAD ON DOORS The load on the doors is given by the water pressure in lbs per sq in multiplied by the area in square inches. The water head will naturally be greater at the bottom

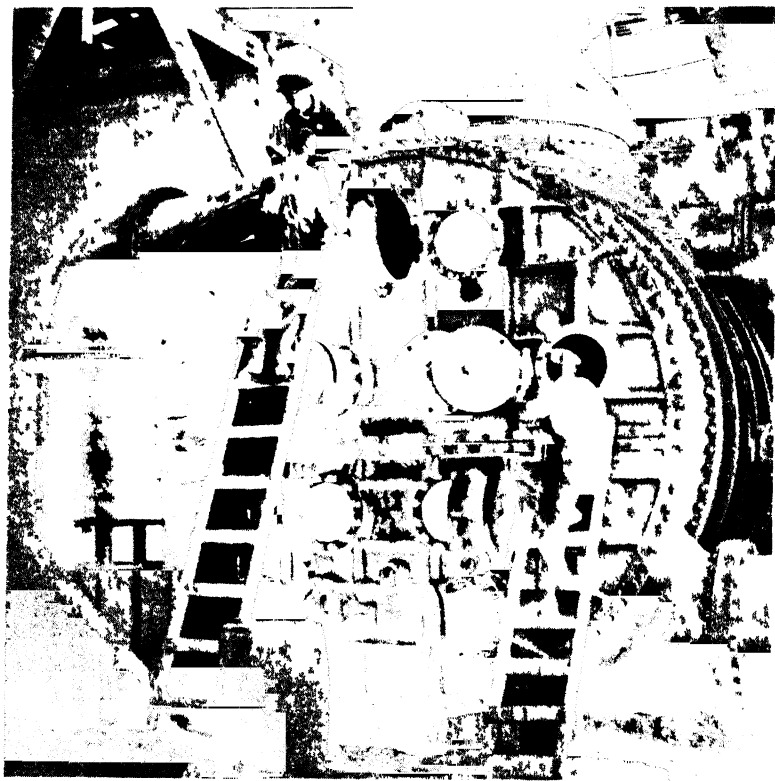


FIG. 24 (see p 71)
AILES CHAIMERS CONDENSER SHOWING HINGED INSPECTION
DOORS

than at the top, as, apart from the difference in static head, the bottom section of the door adjoining the water inlet branch will be subjected to the full pressure due to the friction head in the condenser. Where the pump delivers the water through the condenser to the top of a cooling tower, the maximum head will occasionally exceed the 30 lbs per sq in pressure, which is generally taken as the test pressure.

For such conditions, a test pressure of 50 lbs. is usually specified. Since the doors have to stand the test pressure which is uniform over the whole area, the designer will have to base the strength on this pressure and not on the variable working load.

The exact calculation of the thickness of the metal required to stand the loading* is hardly practicable, since the doors are ribbed or reinforced by T irons or by means of stay-bolts.

* See C. C. Poinder, "The Strength of Dished Ends"—*The Association of Engineering and Shipbuilding Draughtsmen*, Session 1923-24.

CHAPTER X

SURFACE CONDENSERS—TYPICAL DESIGNS

Contraflo Condensers. The steel shell condenser shown in Fig. 26 was built by Messrs. Richardsons-Westgarth, Ltd., of Hartlepool, who, together with many other manufacturers in this country and on the Continent, build condensers under licence from and to the designs of Messrs. The Contraflo Condenser Co., of Westminster. The condenser shown has a cooling surface area of 11,500 sq. ft., and is designed to condense 86,000 lbs. of steam per hour at a vacuum of 27.75 ins. when supplied with cooling water at a temperature of 82° F.

The following table gives some interesting constructional details of this condenser—

Length between plates	15 ft .
Overall diameter of tube-plate	7 ft 11½ ins
Overall height	10 ft. 2 ins
Thickness of tube-plate	1½ ins.
Thickness of shell plating	½ in
Shell stiffening	4 ins × 3 ins × ⅝ in angles 6 ft × 4 ft. × ⅝ in. tee bars
Riveting	⅞ in diameter, 2½ ins. pitch
Steam opening	circular 5 ft. 7 ins. diameter
Total weight (empty)	36 tons

The diagrammatic view of a tube plate shown in Fig. 27 can be taken as representative of the Contraflo Co.'s standard designs.

PRINCIPLES OF DESIGN. The steam inlet is generally arranged so that it extends fully two-thirds round the condenser, thus providing an initial tube surface of exceptionally large dimensions to meet the incoming steam. This arrangement gives a large area to steam flow between the tubes, with consequent minimum resistance to flow.

The changing relative volumes of steam and air during condensation are fully recognized, and in the region where the steam is predominant, as at the top portion of the condenser, the arrangement of the cooling surface is such as to allow for a maximum rate of condensation, the tube pitching being comparatively wide. As the steam condenses and the proportion of air present becomes more pronounced, the cooling surface of the condenser is divided into two or more parallel compartments according to requirements. These subdivisions are made by plates arranged and shaped so that they lie exactly in line of steam flow through the condenser,

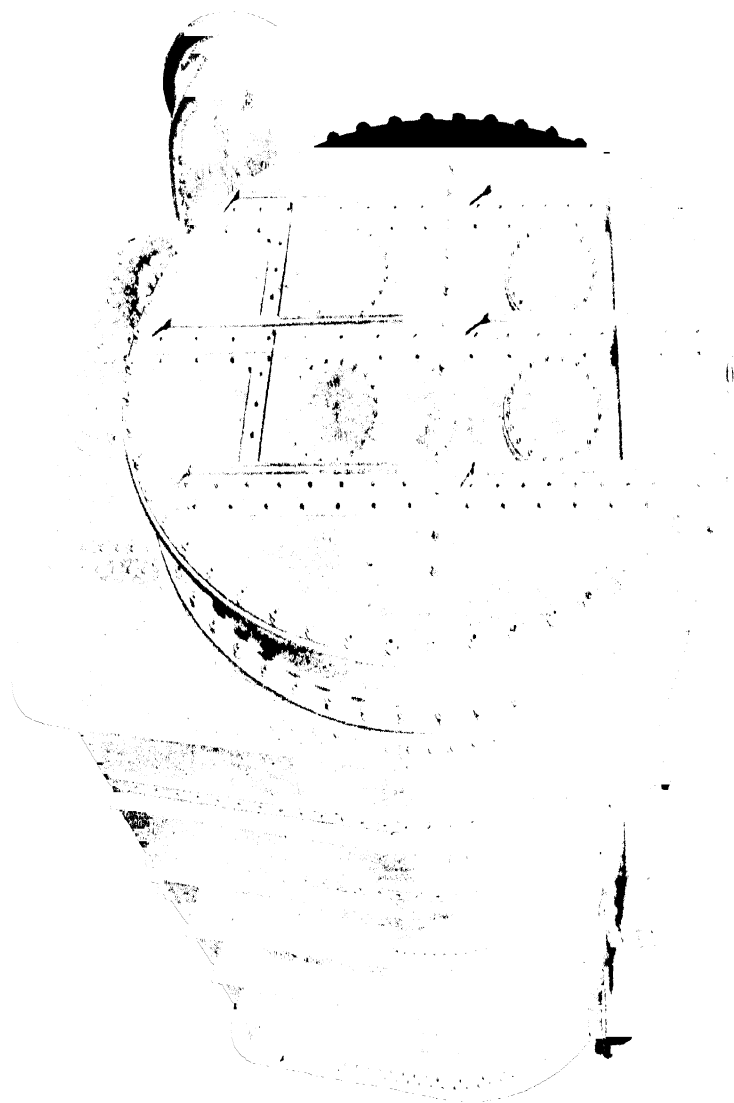


FIG 26 (see p 75).
RICHARDSON'S-WESTGARTH CONTRAFLO STEEL SHELL CONDENSER.

and thus in no way do they interfere with the passage of the steam or cause resistance. These divisions are also arranged so that each forms a wedge-shaped compartment and each is really in itself a separate condenser. The effect of these divisions is such

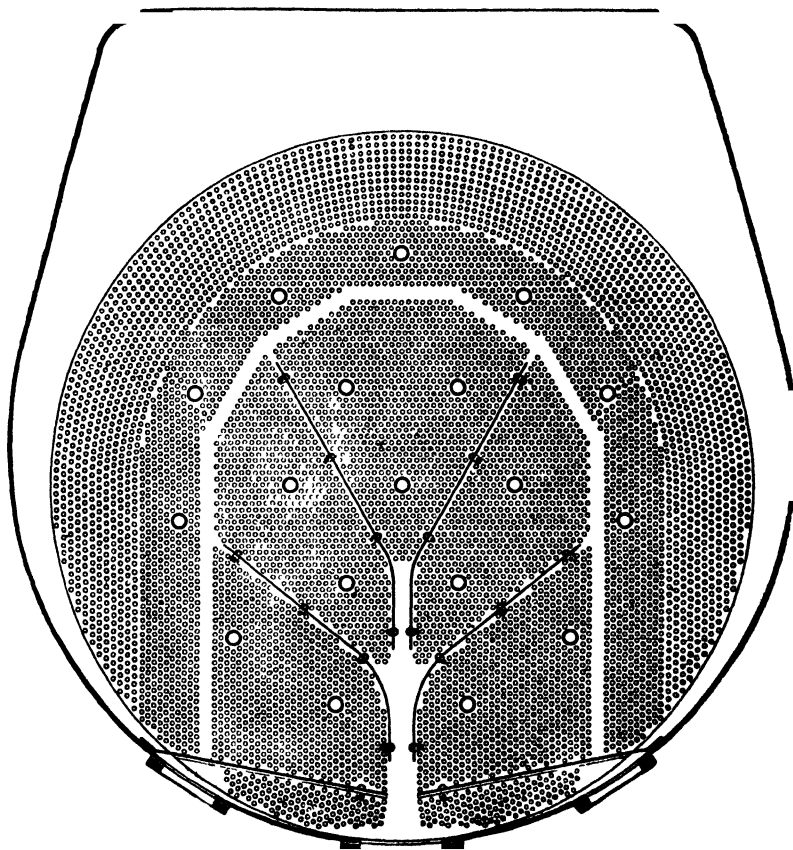


FIG. 27 (see p 75)

DIAGRAMMATIC VIEW OF STANDARD CONTRAFLO DESIGN.

that air and uncondensed gases are gradually concentrated towards the narrow outlet, with the result that the air at that point is of greater weight per unit of volume than at any other part of the condenser, and is consequently in the most desirable condition for withdrawal by the air pump. The outlets or narrow ends of these compartments meet at a point close to the condensate outlet and the devaporizing chamber inlet. In this devaporizing chamber

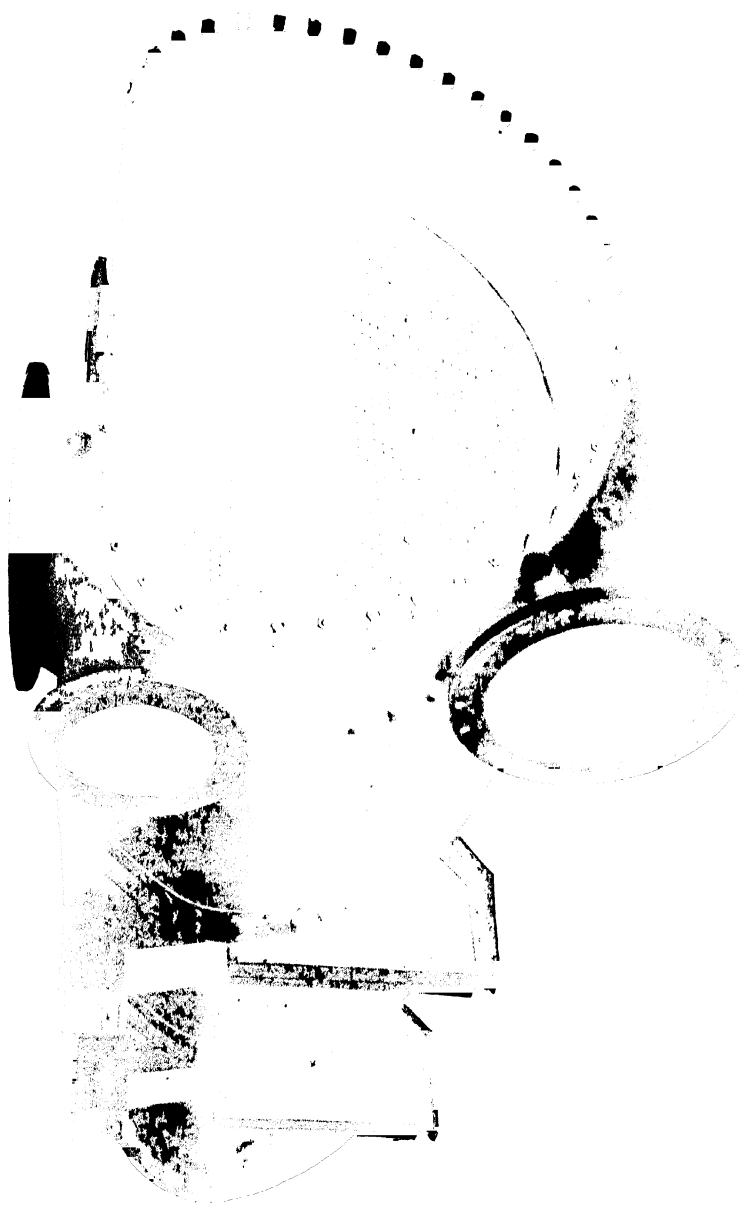


FIG 28
VICKERS CONTRAFLO MARINE CONDENSEP

the quantity of vapour still in association with the air is further reduced by condensation, and the air is cooled down to a low temperature. By this condensation of the vapour and cooling of the air, the volume of aerated vapour to be extracted by the air pump from the condenser is reduced as far as practicable.

It will be recognized that by this method of air contraction the kinetic energy of the inflowing steam at the top of the condenser materially assists in driving the air towards the narrow ends of the compartments, consequently the air region occupies a very small portion of the cooling surface.

A further advantage claimed for this design is the thermal efficiency obtained from the condenser, resulting from the high condensate or hot-well temperature. This is largely due to the

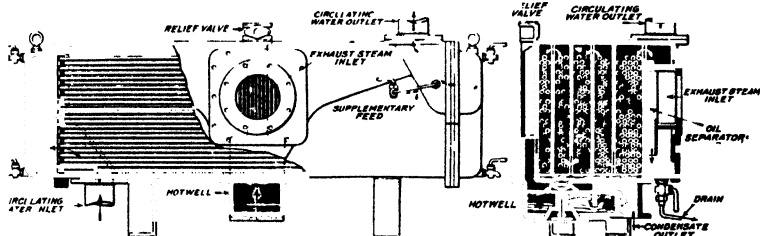


FIG. 29

AUXILIARY CONTRAFLO CONDENSER

condensed steam being caught by the air concentrating plates, which shelter the tubes below and prevent the falling condensate showering over the colder tubes in the lower part of the condenser.

CONTRAFLO MARINE TYPE CONDENSERS. Fig. 28 shows an underslung Marine type Contraflo condenser built by Messrs. Vickers, Ltd., Barrow-in-Furness, having a cooling surface of 6,500 sq. ft., and designed to maintain a vacuum of $28\frac{1}{2}$ ins. with sea at 65° F. The two branches shown at the top of the water box are for circulating water discharge and bilge injection discharge. It will be seen that the shell is of steel, but the water box is made of cast iron.

Owing to the very small space available below turbines on ship-board, underslung condensers are generally of small depth compared with their width, and therefore the length of steam flow over the surface, and consequently the resistance, is reduced. To provide for a large admission area the exhaust inlet from the turbine is carried nearly across the whole width of the condenser.

CONTRAFLO ATMOSPHERIC CONDENSER. Fig. 29 shows an auxiliary atmospheric Contraflo condenser. These condensers are very extensively used on board ship, and the design aims at attaining

a high condensate temperature under all conditions of loading. For this purpose a number of compartments are arranged in series with one another, and the second compartment only comes into operation when the first one is fully loaded, and so on. Since the condensate produced in each compartment is collected at the bottom of the condenser, and is not carried over to later compartments, the condensate is not subjected to the cooling effect due to its coming into contact with a large cooling area after it has been formed. The

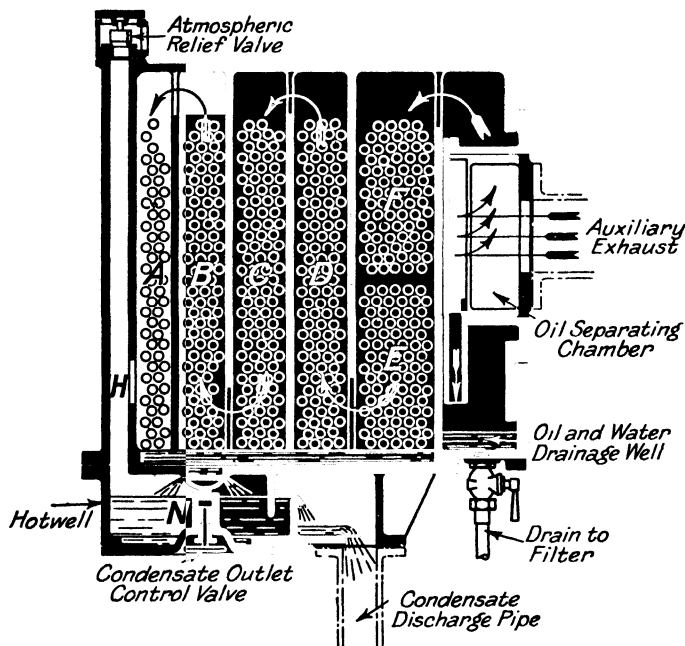


FIG. 30

DIAGRAMMATIC ARRANGEMENT OF CONTRAFLO CONDENSER.

advantage of this arrangement is naturally very pronounced under conditions of light load. Fig. 30 illustrates this type of condenser in diagrammatic form.

The steam, on entering the oil-separating chamber, impinges against a ribbed surface on which the oil is deposited, and from which it falls down through a duct into the drainage well. The cleansed steam passes through openings in the side of the oil separator and enters the condenser, through which it flows in the direction indicated by the arrows, the air escaping through the atmospheric relief valve.

The condensate collects in the base of the condenser and normally flows away at a high temperature through the valve *N*, the level of the condensate in the bottom of the condenser being below the level of the tubes. Should operating conditions arise which make it desirable to reduce the temperature of the condensate, as when the condenser is heavily overloaded and the condensate becomes too hot for pumping, the valve *N* is partially closed, thus causing the level of the condensate to rise and drown a number of the lower rows of tubes, thereby reducing the condensate temperature. Should the maximum amount of cooling ever be required, the valve *N* can be totally closed, and the condensate rises until it flows in its cooled condition through the opening *H*. Under all normal conditions it will be found possible to work with the valve *N* full open, and thus obtain the highest thermal efficiency.

The condenser has five water passes, as follows: First Pass through the two tube nests marked *A* and *B* in parallel; Second Pass through the tube nest marked *C*; Third Pass through the tube nest marked *D*; Fourth Pass through the tube nest marked *E*; Fifth Pass through the tube nest marked *F*.

The oil and water from the drainage well pass away through the drain pipe to the filter tank. By means of the oil separator, oil and grease are prevented from reaching the tubes; thus the condensing surface remains clean for a prolonged period so that the high efficiency of condensation is maintained.

Dirt can be removed from the condenser by removing the hotwell, which covers an opening extending the full width of the condenser.*

The Weir Uniflux Condenser. Messrs. G. & J. Weir, Ltd., of Cathcart, Glasgow, have specialized largely in the development of condensers for marine turbine use, the design being known as the Weir Uniflux Condenser.

The most noteworthy feature of this design is the shape as illustrated in Fig. 31. The object of adopting the inverted pear-shape construction is to obtain practically uniform steam velocity throughout the passage across the surface without the use of baffle plates or partitions. Provision is made in the bottom of the condenser for ensuring equality of distribution so that no short circuiting of any portion of the surface can take place. It is claimed that the directness of the flow eliminates the risk of ineffective zones being present. These condensers are usually designed to

* Some interesting test results obtained on auxiliary condensers of this and other types were published in a Report to the Council of North East Coast Institute of Engineers and Shipbuilders by Messrs. C. Waldie Cairns, M.Sc., and J. Morrow, M.Sc., D.Eng., Session 1919-1920.

operate with a high water velocity to ensure a high heat transmission coefficient.

The shells are generally made of steel plate. In the design shown in our illustration, the water box and door are formed in one casting

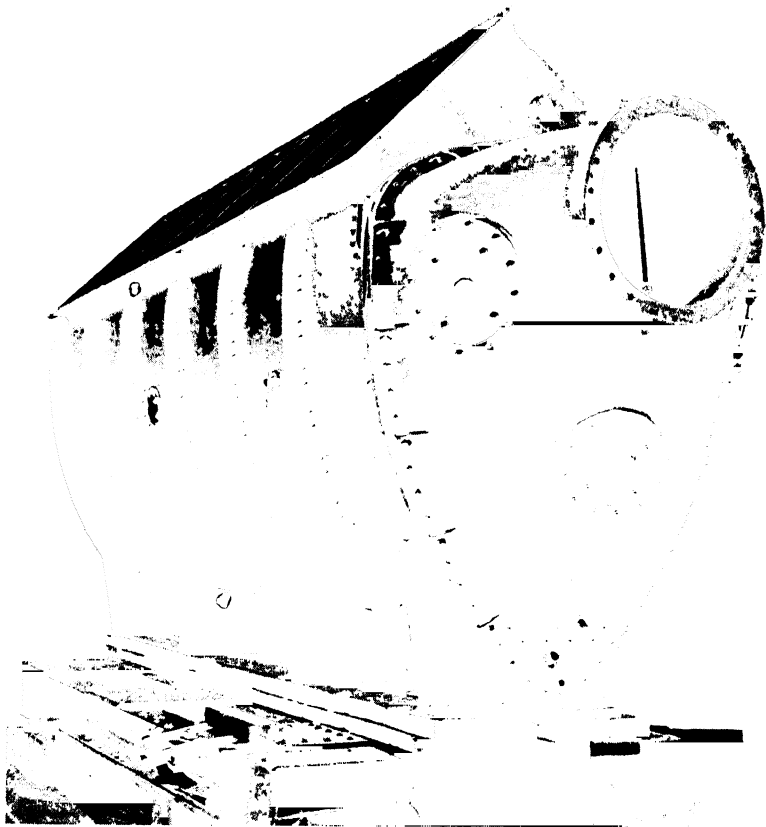


FIG. 31.

WEIR UNIFLUX MARINE CONDENSER.

with the water branch at right angles to the axis of the condenser. It will be apparent that a very low friction loss can be secured at the branches with the design shown. Where working with reciprocating engines, Messrs. Weir recommend a vacuum of 25 ins. to 26 ins. with cooling water at 65° F., except in the case of vessels trading in cold latitudes, where a vacuum of 26 ins. with 60° F. sea water is considered preferable.

In the case of turbine marine vessels a vacuum of 28½ ins. is

suggested for normal sea water temperatures up to 60° F., 28 ins. for 75° F., and of 27½ ins. for sea water at 85° F.

WEIR AUXILIARY CONDENSERS. Messrs. Weir have developed a complete series of standard auxiliary condensers designed to work with minimum attention. Special means have been adopted to avoid the formation of scale in the interior of the tubes, and the hotwell temperature can be regulated to give the highest possible feed temperature.

The condensers have cast iron shells, rolled brass tube plates and ¾ in. tubes, No. 18 gauge. Provision has been made for intercepting oil and grease, which are carried over by the auxiliary exhaust steam in a special filter chamber which forms an integral part of the main shell casting. The filtering material consists of coco-nut fibre arranged in a steel frame which can be easily withdrawn for cleaning purposes. The condensers are fitted with the usual accessories such as soda cocks, scum valve, drain valve and air cock on water box. Table VII gives the leading particulars of the makers' standard designs together with the normal steam duty when working at atmospheric pressure, and at 20 ins. vacuum, based on a water temperature of 90° F.

TABLE VII
PARTICULARS OF WEIR UNIFLUX AUXILIARY CONDENSER
Suitable for an Inlet Temperature of 90° F.

No.	Cooling Surface in sq. ft.	Lbs. Steam per Hr. at Atmo spheric Pressure.	Lbs. Steam per Hr. at 20" Vacuum.	Gallons of Circulating Water Required per Hour.	Exhaust Steam Inlet.	Circulating Pump Required.	Circulating Pump Branches			
							Suc-tion	Dis-charge.	Steam.	Exh't.
1	125	6,000	3,500	160	6½"	7" ^ 6½" x 15"	3½"	3"	1½"	1½"
2	200	9,500	5,500	250	8"	8" ^ 7" x 18"	4"	3½"	1½"	2"
3	300	14,500	8,500	400	9½"	10" ^ 9" x 21"	5"	4½"	1½"	2½"
4	425	20,000	12,000	550	11½"	12" 10½" x 21"	6"	5½"	1½"	2½"
5	550	26,000	15,500	700	13"	12½" 10½" x 24"	6½"	6"	1½"	2½"
6	750	33,000	20,000	890	15"	14" 12½" x 24"	8"	7"	2"	2½"
7	1,000	43,000	26,000	1,150	18"	Centrifugal	8"	8"		1½"

Worthington-Simpson Condensers. Messrs. Worthington-Simpson, Ltd., of London and Newark, standardize on a rectangular shell section in all sizes up to 14,000 sq. ft. They adopt cylindrical shells beyond that size and up to 40,000 sq. ft. Their equipment enables them to handle sizes in one unit up to about 70,000 sq. ft. For smaller sizes it is claimed that the adoption of the rectangular section enables them to cover a large range of sizes with very few patterns. These frames enable them to build condensers in stages

of 200 sq. ft. from the smallest to 4,000 sq. ft., and in 500 sq. ft. stages above this and up to 14,000 sq. ft. The cross sectional area is always based upon an approximate ratio of height to width of 6 to 4, and they have found that by adhering to this ratio they are able to obtain an excellent steam distribution without any dead pockets.

A tube plate arrangement of a condenser of this type has already been considered in Fig. 14.

Fig. 32 shows details of a 20,000 sq. ft. cast iron cylindrical condenser built by Messrs. Worthington-Simpson for the Lancashire Electric Power Co., designed to maintain a vacuum of 28.5 ins. when dealing with 130,000 lbs. of steam per hour, exhausted from a 10,000 kw. B.T.H. turbine when supplied with cooling water at 65° F.* The auxiliaries are in duplicate throughout, and are so proportioned that the full steam conditions are met when both sets of auxiliaries are in use and the full vacuum is maintained at 60 per cent of full load when only one set of auxiliaries is kept running.

The condenser shell is built up in four sections bolted together across the horizontal and vertical centre planes. The condenser has two water passes and only one waterway is used. The general details of constructions and leading dimensions are clearly indicated on the drawings. The tube plates are 1½ ins. thick, and the tubes are ¾ in. outside diameter, No. 18 S.W.G.

Worthington Pump and Machinery Corporation. Fig. 33 shows a 56,000 sq. ft. condenser built by the Worthington Pump & Machinery Corporation, of New York. The tube plate and tubes are placed eccentrically in the shell so that a steam belt is formed round a considerable portion of the nest of tubes which, in addition to the vertical steam lanes indicated by the undrilled portions of the tube plate, ensure efficient distribution of the steam. This company claim to have originated the internal air cooler which can be seen at right-hand bottom section of the condenser. The cooler section is separated from the main body of the condenser by a steel baffle plate extending the full length of the shell, and in its upper portion tapering inwards to about a quarter of the full length at the bottom where it is supported from the bottom of the shell. The tapered edges are bent upwards so that the plate forms a tray and the whole of the condensate that drops on to the plate is discharged over the short bottom edge that extends immediately over the condensate outlet. The turned-up edges prevent a shower of condensate pouring into the cooler space.

* See pages 338-340; also *Engineering*, 22nd and 29th Dec., 1922.

The illustration shows the stay-rod which extend through the flat doors. The intermediate tube support plates are made of cast



FIG. 33 (see p. 85).

WORTHINGTON PUMP AND MACHINERY CORPORATION, 56,000 SQ. FT.
CONDENSER.

iron and are generally so spaced as to allow not more than 6 ft. of unsupported tube.

The Wheeler Condenser & Engineering Co. The following 45,000 sq. ft. condenser built by the Wheeler Condenser & Engineering Co. of Carteret, New Jersey, is installed at the Seal Branch Station of the Los Angeles Gas and Electric Co., California.

This condenser serves a 30,000 kw. turbine, having an overload capacity of 35,000 kw.* The authors are indebted to the Geo. H. Gibson Co., of New York, for the following information relating to this plant.

The condenser is designed to maintain a vacuum of 28.96 ins. when dealing with 380,000 lbs. of steam per hour, and when supplied with 50,000 gallons of cooling water per minute at 55° F. The water circuit is divided vertically into two sections, and each section is arranged for two water passes. The auxiliary pumps are arranged in duplicate, each set serving one section of the condenser so that half the condenser can be shut down for cleaning purposes in the manner described in Chapter XI. The condenser is provided with baffle plates on either side that protect the air suction from dropping condensate. The shell is of shallow design and the tube plates are arranged with numerous steam lanes to ensure a satisfactory distribution of the steam and to reduce the length of the steam path. The condenser shell is built up of cast iron sections in accordance with the makers' usual practice. The weight of this condenser empty is 180 tons (British), and it contains 8,186 1-in. tubes No. 16 B.W.G. The leading dimensions are—

Height from centre line of condenser to centre line of water outlet branch	4 ft. 3 ins.
Height from centre line of condenser to top of exhaust connection	13 ft. 4½ ins.
Height from bottom of hot well to centre line of condenser	8 ft. 6½ ins.
Size of exhaust opening	11 ft. 6 ins by 20 ft.
Length of tubes	21 ft.
Maximum width of condenser	17 ft. 8 ins.

The company invariably use cast iron shells for land work, but adopt steel or copper shells for marine purposes. Special attention has been given to the spacing of the tubes and this, together with the introduction of suitable steam lanes, ensures proper distribution of the steam.

The company have equipped a special tube works for the production of the whole of the tubes required in the manufacture of their condensers, evaporators and feed heaters. It is claimed, and no doubt with some justification, that they are better able to ensure securing the desired quality of tubes by having their complete manufacture under their own control.

UNDERTYPE WHEELER CONDENSER. Fig. 34 gives a general view of the company's well-known condensers mounted directly over combined direct driven air and circulating pumps suitable for

* For a description of this station see *Power*, 2nd June, 1925.

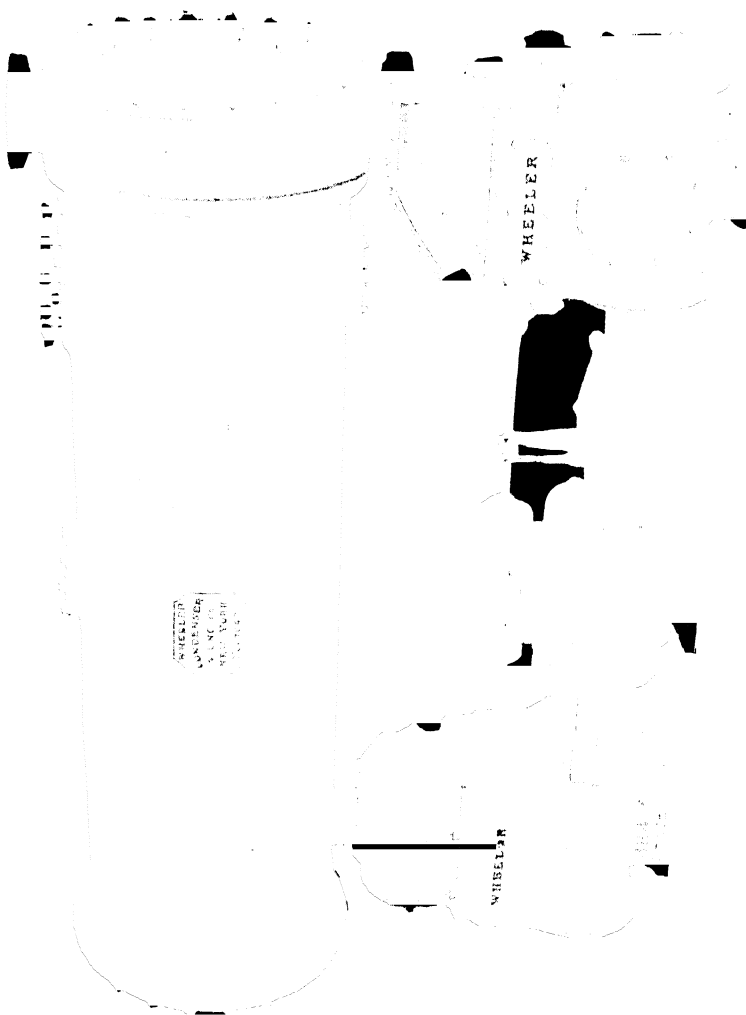


Fig. 34 (see p 87).
WHEELER UNDER-TYPE CONDENSER

vacua up to 26 ins. Latterly, the company have developed a somewhat similar compact plant in which the reciprocating pumps are superseded by turbine-driven centrifugal circulating water and condensate pumps, combined with a single stage steam air ejector.

WHEELER WATERWORKS CONDENSERS. Fig. 35 shows one of the company's waterworks condensers in which the exhaust steam passes through the tubes and circulating water passes externally, so that the whole of the water delivered by the main pumps can be

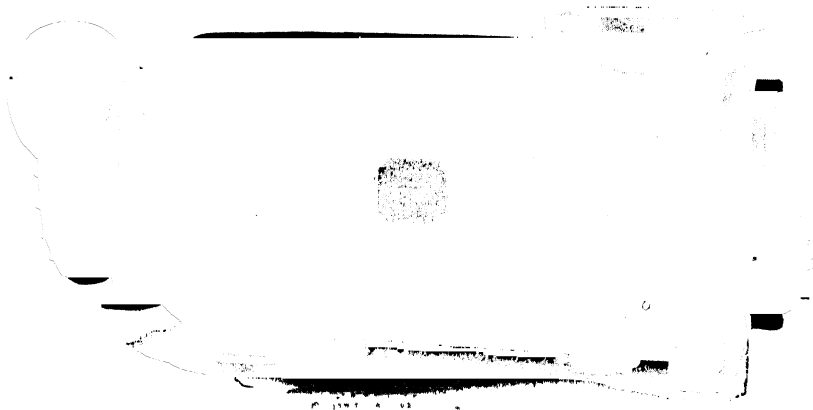


FIG. 35.
WHEELER WATERWORKS CONDENSER.

passed through the condenser without undue friction loss. Naturally such condensers are primarily adapted for moderate vacua owing to the restricted steam space. For high-vacuum and large waterworks engines generally, the company recommend the use of their standard condenser designs by passing only the required water quantity through the condenser.

Alberger Spiroflo Condensers. The surface condensers built by the Alberger Pump & Condenser Co., of New York, and named Spiroflo Condensers, aim at embodying in a cylindrical shell the principle of a triangular cross section, in which the steam is admitted over one side and the surface diminishes with the reduction in volume towards the apex at which point the air is extracted. (Fig. 36.)

The construction of the steam dome which stretches over practically the whole length of the condenser, and whose sides are tangential

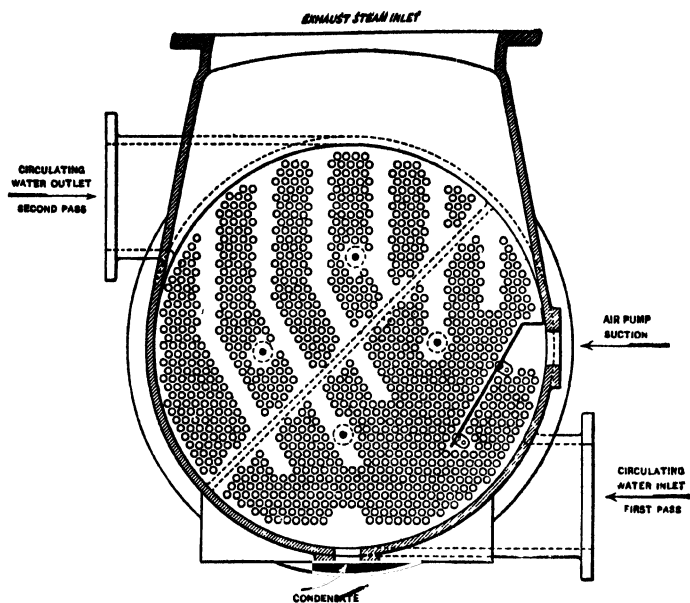


FIG. 36.
ALBERGER SPIROFLO SURFACE CONDENSER.

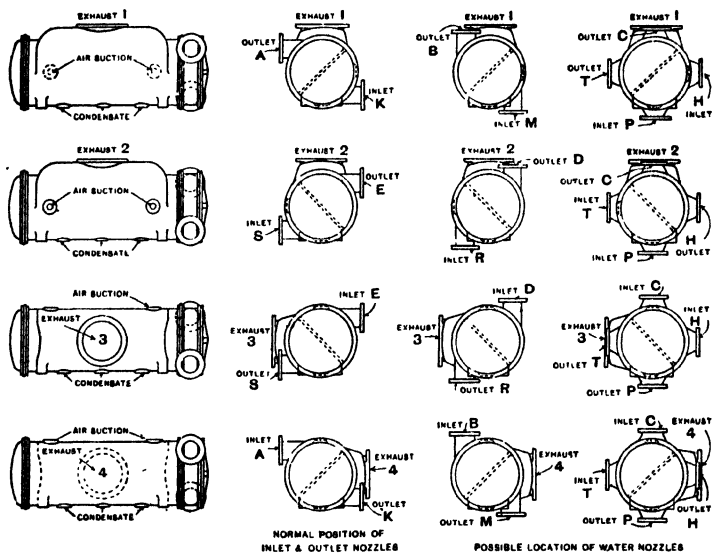


FIG. 37.
STANDARD WATER BRANCH POSITIONS FOR ALBERGER CONDENSERS.

to the shell, exposes a substantial tube area to the incoming steam. The effective distribution of the steam is further assisted by the numerous deep steam lanes reaching towards the air cooler section and the air outlet.

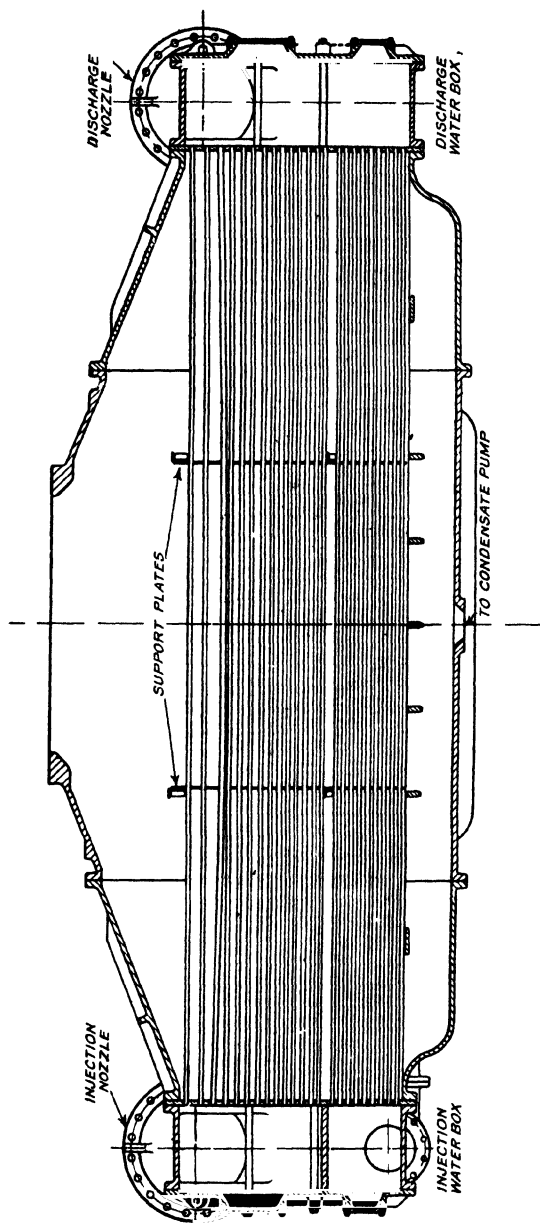
All condensers built by this firm are of the two-pass type, with the water division plate at an angle of 45° . The advantage of this construction is not at first apparent, but it allows for many convenient arrangements of the exhaust openings and of the water branches without sacrificing the correct counter-current principle. The various standardized positions for these openings are shown in Fig. 37. In all cases the condensate is extracted from the bottom through three openings distributed over the bottom centre line.

The Ingersoll Rand Condensers. Fig. 38 shows a longitudinal section through a condenser of Messrs. The Ingersoll Rand Co.'s manufacture. The steam is admitted into a large chamber designed to ensure complete distribution of the steam over the entire length. The steam flow to the ends of the condenser is further assisted where necessary by eccentric drilling for a few tubes in the support plates, but not in the main tube plates, for the purpose of adding a slight compensating resistance to offset the influx velocity of the steam. The spacing of the tubes is kept very wide at the top and diminishes gradually towards the bottom. The shell is wedge-shaped and this, combined with the diminishing tube spacing towards the bottom, is claimed to give a practically even steam velocity throughout the condenser, the reduction in steam space corresponding to the reduction in volume. With this design, no steam lanes are required.

EXTERNAL COOLER. The air passes over an external cooler before reaching the steam jet which acts as first stage of the air pump. This cooler is illustrated in Fig. 39, and is shown in section in Fig. 40.

The external cooler shell is of cast iron and contains cast iron grids, through which cooling water is circulated. The grids are secured to the cooler water heads by accurately machined, cast iron nipples pressed into place. Large plugged holes are provided in the shell opposite each grid for inspection and cleaning of the water passages. Flanged openings are provided on the cooler shell for water inlet, water outlet and air outlet. Where steam jets are used, the primary jets are mounted on the cooler and discharge into an intercondenser forming part of the main cooler.

It may here be noted that this type of air cooler complies with the condition suggested on page 41.



I 1 38 sep 91)
SECTION THROUGH INGERSOLL RAND SURFACE CONDENSER

INGERSOLL RAND TUBE SPECIFICATION. The company carry out rigid tests on condenser tubes, and their specification is of interest and reads as follows—

CHEMICAL REQUIREMENTS. Test tubes are taken from each lot of 5,000 lbs. or less and, when chemical analysis is made, equal quantities of millings from

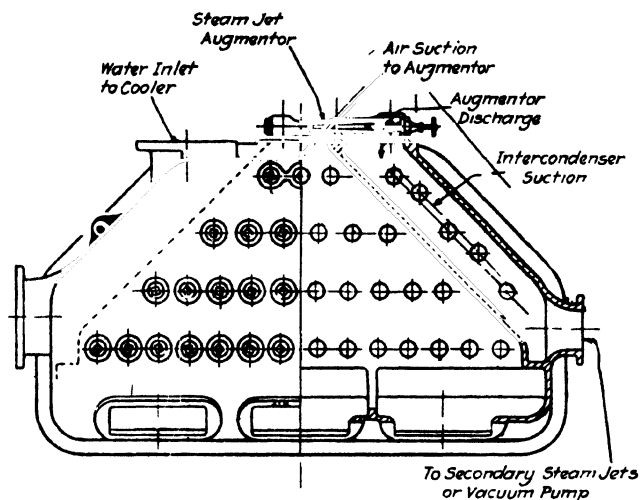


FIG. 39 (see p 91).
INGERSOLL RAND AIR COOLER.

at least four separate tubes are thoroughly mixed to form the sample to be analysed. The chemical requirements are listed below—

Admiralty. Copper, 70 per cent minimum ; tin, 1 per cent minimum ; lead, .075 per cent maximum ; iron, .06 per cent maximum ; zinc, remainder.

Muntz. Copper, 59 per cent minimum ; lead, .6 per cent maximum ; iron, .1 per cent maximum ; zinc, remainder.

Copper. Copper and zinc 99.88 per cent minimum.

PHYSICAL REQUIREMENTS. All test tubes must pass the hammer and pin tests. The tubes are hammered flat on different elements throughout their length until a micrometer caliper, set at three times the gauge, will pass freely over the tube. Pins, tapered $1\frac{1}{2}$ in. to the foot, are then driven into the tube ends until the internal diameter has been increased by 15 per cent for Admiralty metal, 12 per cent for Muntz, and 5 per cent for copper. The tube must pass these tests without sign of fracture. Admiralty and Muntz tubes must, furthermore, withstand an internal hydrostatic pressure of 1,000 lbs. per square inch. If any tube fails in one of these physical tests, two more are selected from the lot and tested in its place.

METALLOGRAPHIC REQUIREMENTS. Brass tubes undergo a metallographic test to determine whether they have been properly annealed. Admiralty

tubes, when properly prepared for this test and magnified to 75 diameters, must show uniform and complete recrystallization with the average crystal size as magnified, not exceeding one-quarter of an inch. Muntz metal tubes, when magnified 75 times, must show uniform and complete recrystallization in the alpha constituent.

FINISH. Good workmanship and smooth finish are ensured by requiring that all tubes be free from material defects on the interior or exterior, well and thoroughly cleaned, true to exterior and interior dimensions, of uniform thickness throughout and carefully straightened.

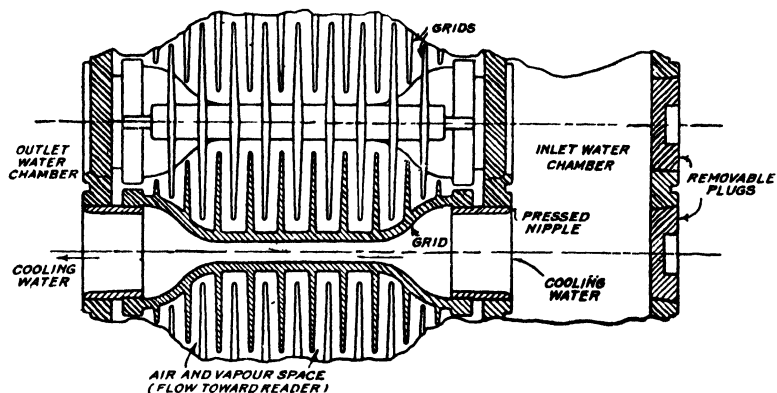


FIG. 40 (see p 91).

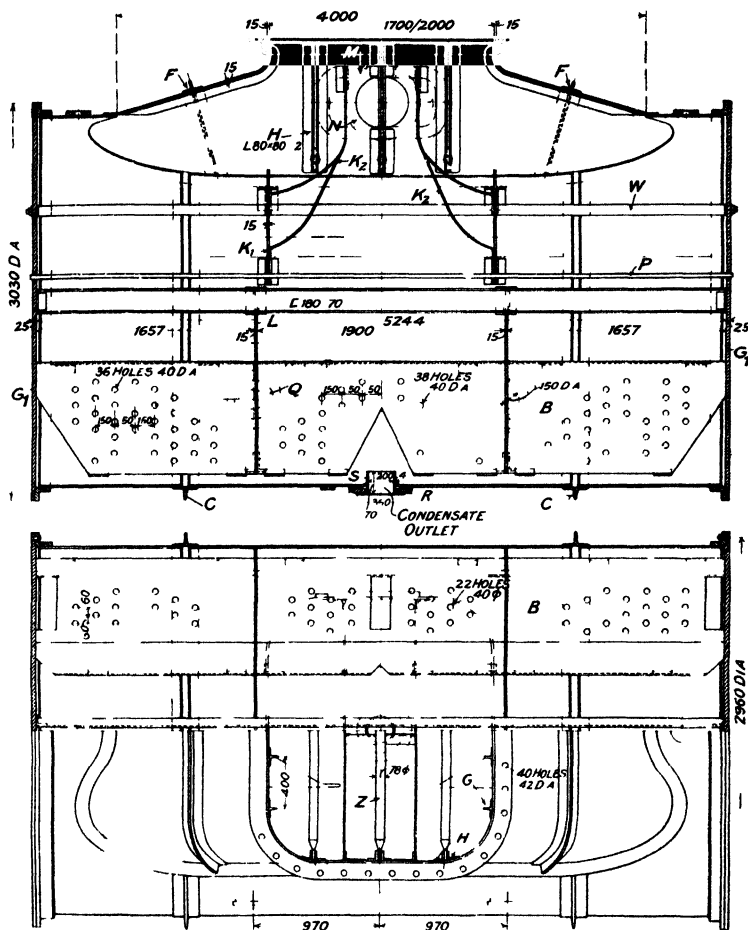
SECTION THROUGH INGERSOLL RAND COOLER

The tubes are expanded into the tube plate with a chamfered entrance on the inlet side, and are secured with the usual type of ferrule and packing at the outlet end.

Escher Wyss Condensers. Figs. 41 and 42 show the details of construction of a steel shell condenser, having a cooling surface area of 15,000 sq. ft., and built by Messrs. Escher Wyss, of Zurich, Switzerland. The main body of the shell is built up of 13 mm. plates reinforced by means of two 200×100 mm. T irons marked C, the end flanges and the main exhaust steam inlet flange consisting of angle irons $130 \times 130 \times 40 \times 30$ mm. The dome forming the connection from the inlet flange to the body of the shell is made of 15 mm. plate strengthened by T irons, marked F. 160×30 mm. The dome is supported internally by 140×70 mm. T irons, marked G, and by 80×80 mm. angle irons, marked H. Additional support is provided by the tubular stays Z which stretch across the opening at the maximum width.

The 6 mm. plates marked K which extend from the inlet into the steam space, act as baffle plates, and are intended to distribute the steam across the full length of the condenser. These plates

are screwed internally to the steel tube support plates K_1 which extend half way down the condenser. The tubes are supported in the lower half by similar plates marked L . The tube plates G_1



I ic 41
I SCHIR WYSS 15 000 SQ FI CONDINER

consist of 25 mm steel plates. One plate is riveted to the shell, the joint being made with hemp and red lead, and the other is held in position by counter-sunk bolts and nuts with rubber jointing. Details of the tube plates are shown in Fig 16, page 63. The plates are supported internally by 10 steel tubular stays, W , 80 mm

outside diameter, 5 mm. thick. The tubes *P* are 25.5 mm. outside diameter, 1.25 mm. thick, and are packed by means of brass ferrules in the usual way.

The condensate flange *R* and the air outlet flange *Q* are of cast iron. A short steel stand pipe *S* is fitted inside the condensate flange to maintain the water level within the condenser. Duplicate air pumps are provided in this instance and the air is withdrawn from both sides of the condenser, two air suction flanges being fitted at each side.

Perforated air baffles *B* are fitted in both sides of the condenser and extend from the sides towards the bottom of the condenser, and cut off the air cooler sections of the cooling surface from the main body of tubes.

The atmospheric relief connection consists of an oblong flange, marked *M*, 370 mm. $\times$ 800 mm., riveted to the side of the dome. The circular flange opposite to this and marked *N*, serves as auxiliary exhaust steam inlet branch.

The condenser has three water passes, and is provided with separate cast iron water boxes and doors at either end. The position of the door stays *P* is shown in the drawing. Each box is provided with four inspection doors.

The Gervembroich Condensers. Fig. 43 shows the details of constructions of a 9,200 sq. ft. four-pass condenser built by Messrs. Maschinenfabrik Gervembroich, Germany. The tube plate layout for this condenser has already been described (see Fig. 15 *ante*). The design is notable in that the steel plate shell and the water boxes are made in one piece. The tube plates are dished 28 mm. thick, and are riveted radially to the body of the shell. The shell itself is built up of 9 mm. plates, the doors and the tube support plate consist of 25 mm. plates.

The left-hand door at the water branch end derives its support by being screwed to the angle irons *A*, fastened to the horizontal dividing plates in the water box. These in turn are supported by vertical steel plates *C*, cut so as to allow the water to pass on either side and through the centre.

The lower part of the illustration shows details of the tubular stays which support the tube plates and the right-hand door.

Brown Boveri O.V. Condensers. Messrs. Brown Boveri & Co., of Baden, Switzerland, build condensers that are characterized by the tube arrangement illustrated in the sectional drawing shown in Fig. 44.

The tubes are placed within a cylindrical shell in two separate groups divided by a V-shaped gap, whose widest opening faces the

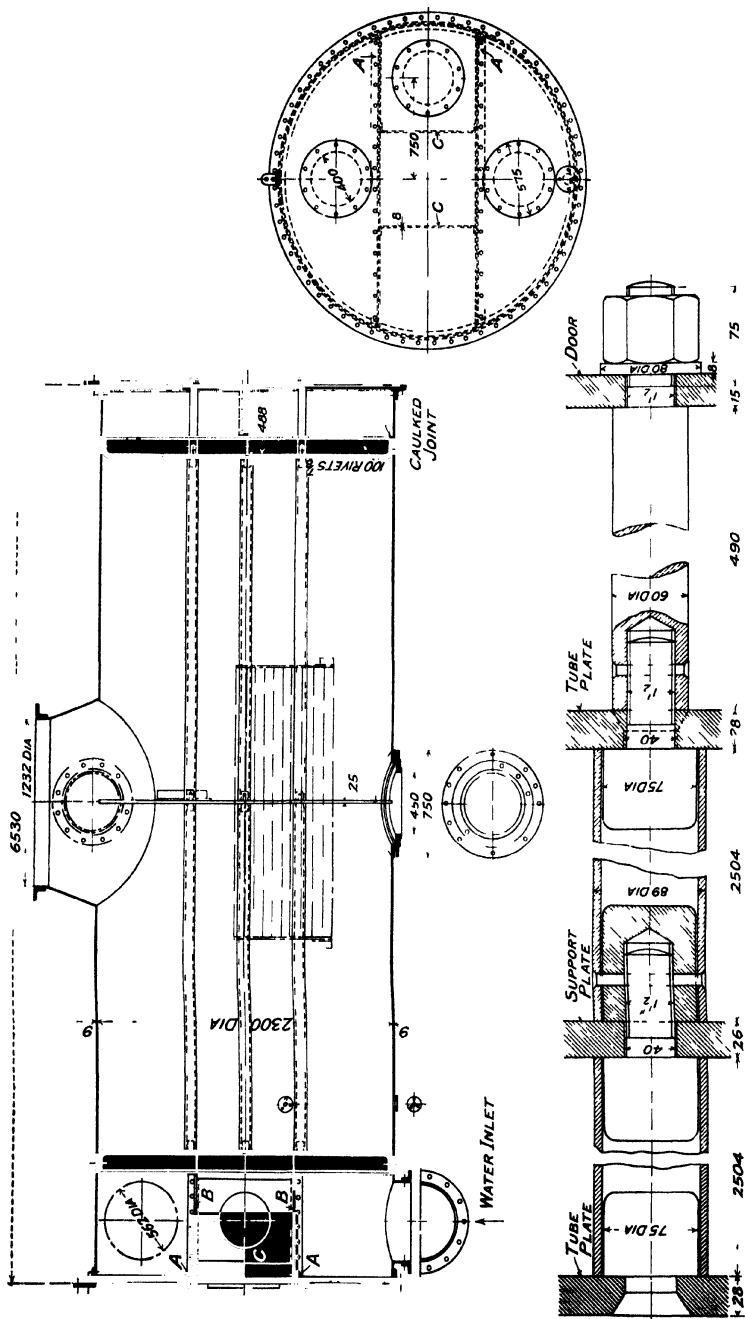


FIG. 43 (see p. 97).
 GERVENBROICH STEEL SHELL CONDENSER AND SUPPORT END

steam inlet and that tapers towards the bottom of the condenser. Since the section of the condenser resembles the letter V within a letter O, the makers have given the design the name of "O.V." condenser.

The steam filling the space between the two groups of tubes is

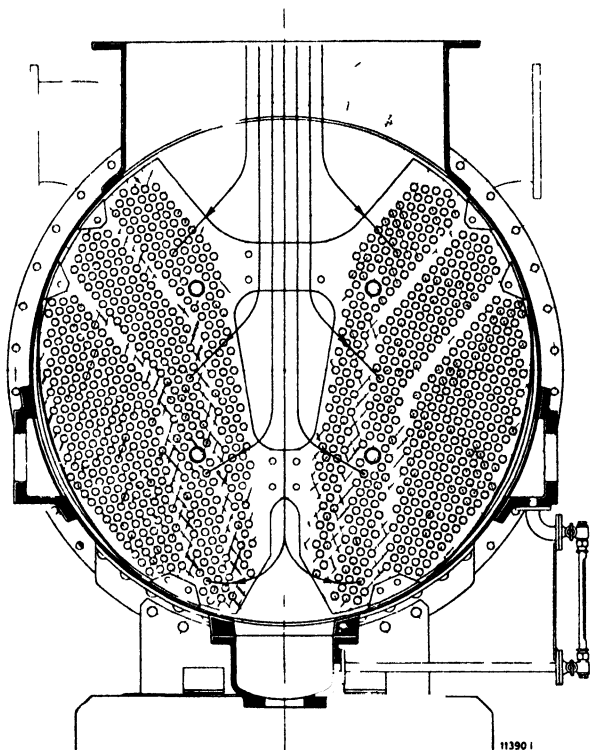


FIG 44

SECTION SHOWING STEAM PATH OF BROWN BOVERI O V CONDENSER

free to distribute itself over the full length of the condenser, and flows outwards across the stacks of tubes towards the two air pump suctions in the manner indicated by the arrows shown in the illustration.

To ensure an equal suction at both air outlet branches, perforated metal sheets are fitted in front of the orifices between the stacks of tubes and the shell.

The cooling water flows in a direction opposed to that of the steam in the manner indicated in Figs. 44 and 45, that is to say,

it flows from both sides towards the axis of the condenser. The water can either be carried away through a single outlet branch, common to both halves of the condenser, or two separate outlet branches can be provided, as in the case of the continuous service condensers built on this principle and described later

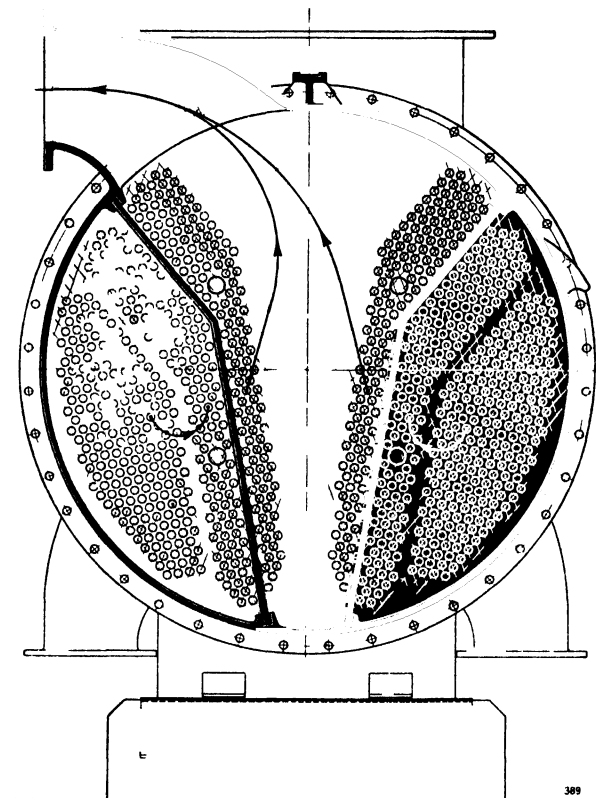


FIG 45

SECTION SHOWING WATER PATH OF BROWN BOVERI O.V. CONDENSER

This design has the advantage that, whilst the convenient cylindrical shape is retained, the steam path across the tubes is exceedingly short and the section of the stacks of tubes across which the steam passes diminishes rapidly towards the air pump suction, so that the steam velocity is maintained. At the same time, the V gap allows the steam ample space to distribute itself over the complete length of the condenser. The provision of such an ample

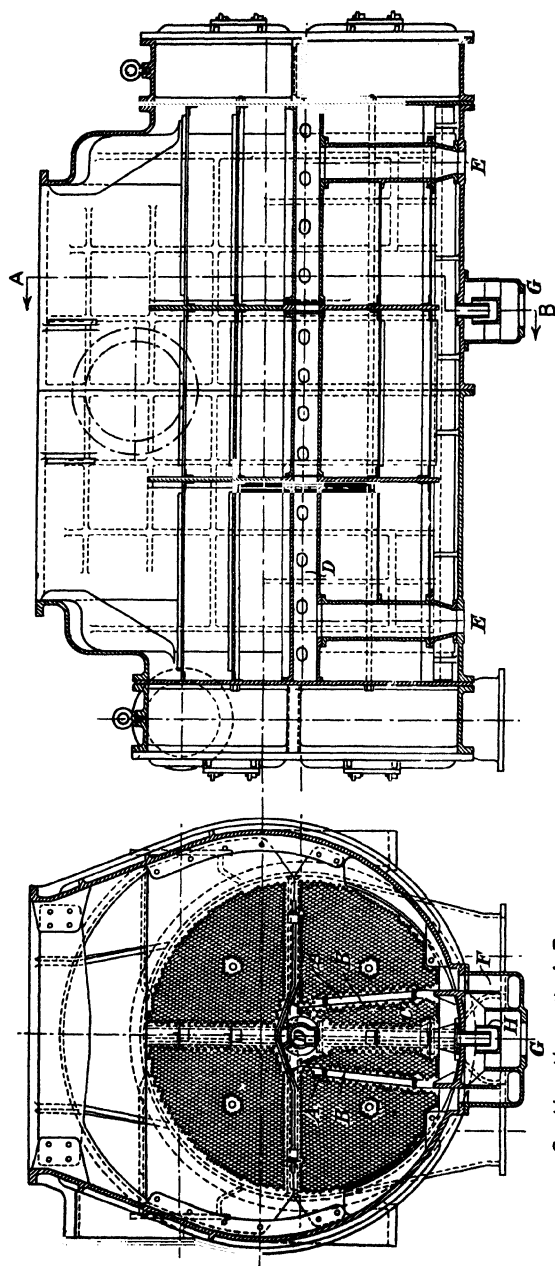
steam space apparently reduces the ratio between the cooling surface and the volume of the shell. In practice, it has been found possible to reduce the spacing of the tubes on account of the shortness of the steam path to such an extent that the same ratio can be maintained as in an ordinary condenser. At the same time, the makers claim an improvement of 20 to 30 per cent in heat transmission coefficient due to the adoption of the present design.

Metropolitan-Vickers Condenser. Messrs. Metropolitan-Vickers Electrical Co.'s experience in the manufacture of surface condensers led them to the conclusion that the usual form of construction and arrangement of tubes was not well adapted to large units designed for high vacua. They experienced considerable difficulty in obtaining with such condensers a reasonably high heat transmission coefficient coupled with a sufficiently low drop in steam pressure across the condenser. A long series of experiments showed that the drop in pressure amounted in some cases to 0.5 in., whereas in smaller condensers it had been of the order of 0.1 in. The reason for the excessive drop was, of course, to be sought in the length and nature of the steam path from the point of inlet to that of outlet. The introduction of steam lanes of varying shape failed to supply an adequate solution to the problem, and they concluded that the shape and size of steam lanes did not influence the result to a material extent.

THE "CENTRAL FLOW" CONDENSER. A solution to the difficulty was discovered by the Metropolitan-Vickers Electrical Co. in their "Central Flow" condenser, the principle of which is illustrated diagrammatically Fig. 46.

It will be seen that steam enters the nest of tubes over the whole periphery and flows radially inward towards the centre, the air being drawn off here, instead of at the bottom as in the old straight down flow type of condenser. The steam has therefore only to traverse the comparatively short distance from the periphery to the centre, instead of the full length of the nest of tubes, as was the case with the older type. The result is a much smaller drop in pressure from air pump suction to steam inlet, and consequently the plant will maintain a higher vacuum at the condenser inlet for a given surface or conversely the same vacuum with a smaller surface.

The steam being admitted all round the periphery of the nest of tubes implies that the steam at the bottom of the condenser is practically at the same temperature as that at the inlet. As the whole of the condensate has to fall through this hot steam, a higher



Section through A.B

FIG. 46 (see p. 101).

METROPOLITAN-VICKERS CENTRAL FLOW CONDENSER.

condensate temperature than usual is obtained. In fact, test results show the condensate temperature to be within one or two degrees F. of the temperature corresponding to the vacuum.

In the older type of condenser the air was drawn off near the bottom of the condenser where the partial air pressure is high, whereas in the central flow condenser the condensate is drawn off where the partial air pressure is low. According to Henry's law, the quantity of air necessary to saturate water depends on the partial air pressure in the atmosphere through which the water is falling. In the ordinary type of condenser the partial air pressure at the point where the condensate is withdrawn is many times greater than is the case of the "Central Flow" type. The amount of air that can be carried away with the condensate can therefore be much greater in the case of the straight flow condenser than in that of the present type. The "Central Flow" condenser is, in fact, an excellent de-aerating device, and test results over long periods have demonstrated that the oxygen content of condensate from a "Central Flow" condenser is less than the minimum that is usually guaranteed by manufacturers of de-aerating plant.

"CENTRAL FLOW" CONDENSER WITH DIVIDED WATER PATHS. If desired, the "Central Flow" condenser can be made with divided water boxes, so that one-half can be cleaned whilst the other is in operation.

Fig. 46 illustrates a "Central Flow" condenser with two water passes and with divided water paths, so that one-half may be cleaned while the other half is in operation. It will be seen that the construction of the shell and the arrangement of the tubes are designed to leave a volute round the nest of tubes in order that the steam should have ready access round the whole periphery. Since the air is withdrawn from the centre of the nest of tubes, the steam flows radially from the periphery to the centre. The air and undensated vapour pass from the centre between the baffles *AB* to the bottom of the condenser, and then upwards over a nest of tubes *C*, into the horizontal omnibus main *D*, which extends the full length between the tube plates. Two short vertical pipes connect the omnibus main *D* to the air extraction flanges *E* at the bottom of the condenser.

The condensate is collected at the bottom of the condenser and passed into the seal chamber *F* away to the extraction pump at the flange *G*. Any vapour condensed in the air cooling chamber *C* is drained away through the water seal *H* to the common discharge *G*.

The objects of these seals are as follows—

The extraction pump suction is sealed from the steam volute in

the main condenser in order to prevent the direct flow of steam into the extraction pump. The air cooling chamber C is sealed from the extraction pump in order to prevent re-evaporation of the condensate resulting from the fact that the pressure in the chamber C

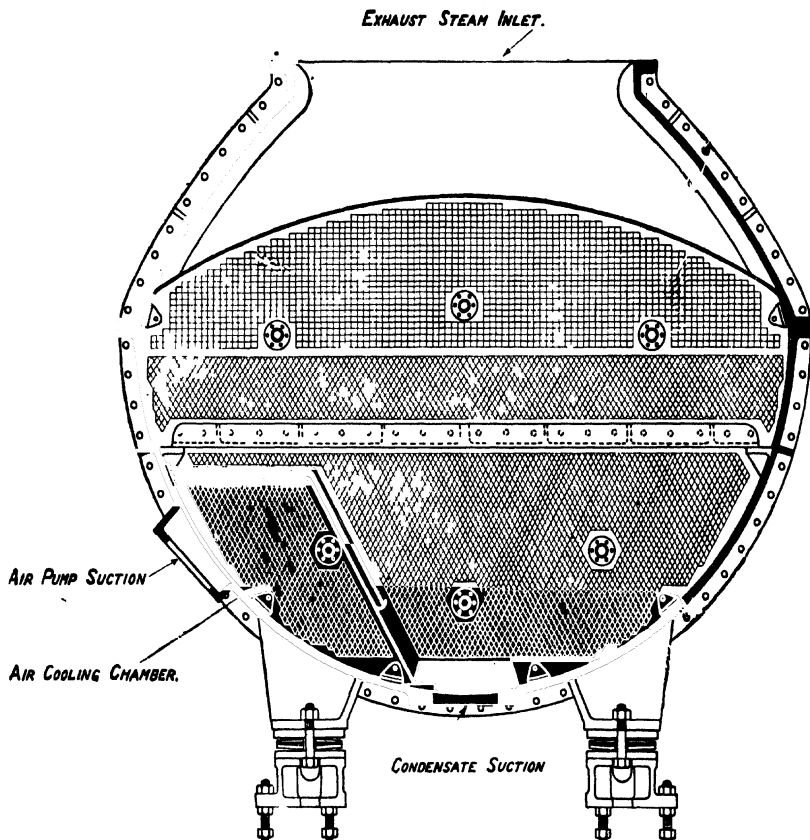


FIG 47

MIRRELL'S WALSON BARREL SHAPED SURFACE CONDENSER

is slightly lower than the pressure corresponding to the temperature of the condensate. This, in turn, results from the fact that the condensate is at practically the same temperature as the steam leaving the turbine.

Compared with the ordinary type of condenser, it will be seen that the periphery over which steam enters the nest of tubes is

practically double in the case of the central flow condenser. Further, the length of the steam path across the nest of tubes, i.e. the number of tubes passed, is reduced to about one half.

Mirrlees-Watson Condensers. Messrs. Mirrlees-Watson Co., Ltd., of Glasgow, have lately departed from their standard circular design of condenser for large units, and have adopted the broad shallow boat-shaped construction shown in Fig. 47.

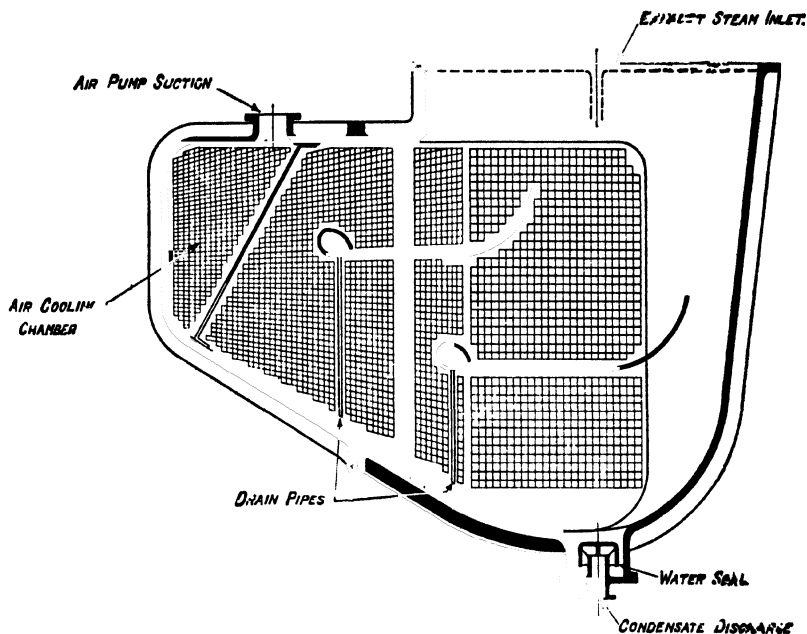


FIG. 48 (see p. 106)

MIRRLEES-WATSON SURFACE CONDENSER HAVING AIR SUCTION ON TOP.

The spacing of the tubes in the upper section is very wide and is reduced gradually towards the bottom. The very ample steam space above the tube nest ensures a uniform distribution of the steam, and the steam path is very short to reduce the drop of pressure across the condenser.

The condenser body is built up of ribbed cast iron sections bolted together, and is supported on springs which allow for the necessary expansion without the need of interposing a flexible expansion piece between the turbine exhaust flange and the condenser. A number of condensers of this type have been installed at the Dalmarnock Power Station of the Glasgow Corporation, and are each designed

to maintain a vacuum of 29.1 ins. when dealing with 160,000 lbs. of steam per hour, and have a cooling surface of 26,000 sq. ft.

Fig. 48 shows another design by The Mirrlees-Watson Co., Ltd., the particular features of which are the position of the condensate and air pump suction branches. It will be noted that the condensate is drawn off at a point close to the entering steam, and therefore at maximum temperature, and the air is drawn off at the extreme end farthest from the steam entrance and condensate branches, and therefore at the coldest point of the condenser.

The horizontal division plates allow for the water condensed by one batch of tubes being collected and led to the bottom of the condenser, without touching the lower batches. This tends towards the highest possible condensate temperature, and reduces the wetness of the lower batches of tubes.

The Société des Condensateurs Delas of Paris and Saint Ouen, who are one of the largest builders of condensers in France, have developed surface condensers embodying novel features due to Mr. M. A. Ginabat, Chief Mechanical Engineer of the French Navy, who has given a detailed description of the principles together with numerous test results in a paper read before the Association Technique de Maritime et Aéronautique of Paris.* The author explains that an increased heat transmission can be looked for by reducing the thickness of the water film on the outside of the tubes. In the usual arrangement of tubes the condensate formed on an individual tube falls on to the top surface of the tube beneath it, and forms a complete film surrounding the whole outer surface of the tube before collecting together with any additional condensate formed at the bottom and falling to the next lower tube, and so on. The whole of the tubes except the top layer are therefore coated with a film of water which interposes an additional resistance between the steam and the metal, and has the effect of increasing the heat transmission resistance R_1 . Mr. Ginabat aims at dividing each tube surface into a wet and a dry portion and by bringing the steam into immediate contact with the dry portion, it is claimed that the overall heat transmission is improved. His object is achieved by staggering the tubes in the manner shown in Fig. 49. It will be seen that the condensate falling from each tube strikes the tube immediately beneath it tangentially and passes round a quadrant of the tube to the bottom, leaving three-quarters of the tube surface dry.

Fig. 50 shows the tube plate layout of a condenser built by the

* *Memoirs de l'Association Technique de Maritime et Aéronautique Paris*, 1924.

Société des Condensateurs Delas under licence from Mr. Ginabat for a 10,000 kw. turbine. The body of the tubes is divided vertically into five sections or tube nests by broad steam lanes which run almost the full depth of the condenser. The nest at the bottom

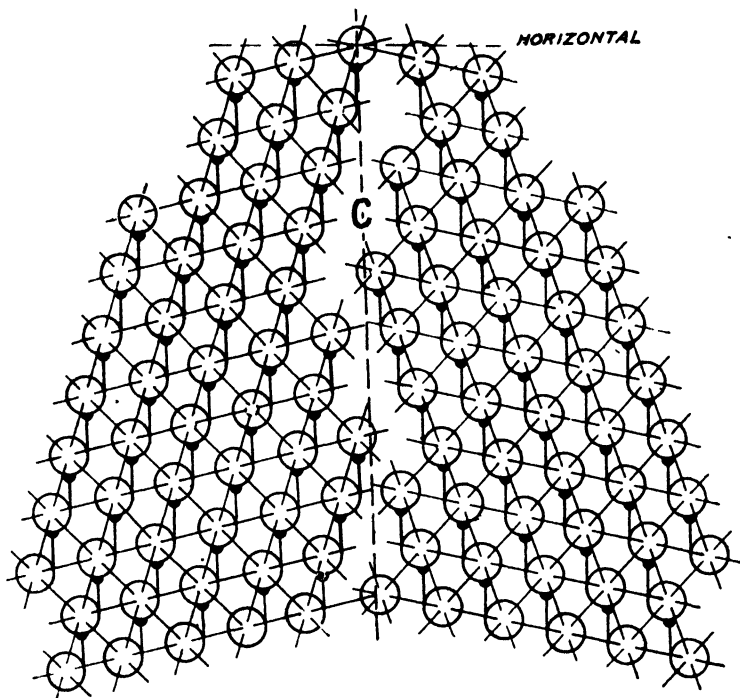


FIG 49.
ARRANGEMENT OF TUBES IN GINABAT CONDENSER.

right-hand side represents the usual arrangement of air cooler. Each of the five nests consists of a right-hand and a left-hand section, as shown in Fig. 49, the arrangement being such that the steam entering the section from the right-hand steam lanes comes in contact with tubes whose left-hand bottom surface is wetted by dropping condensate and vice versa. A vertical gap C is formed between left- and right-hand sections of each nest and the air and uncondensed vapour pass downward through these gaps and over the air cooler section to the air pump connection.

The author claims the following advantages for the present design.

The heat transmission coefficient is raised by reducing the water film on the outer surface that is in contact with the steam.

The short steam paths from the steam lanes to the central gaps *C* ensure a small pressure drop across the condenser.

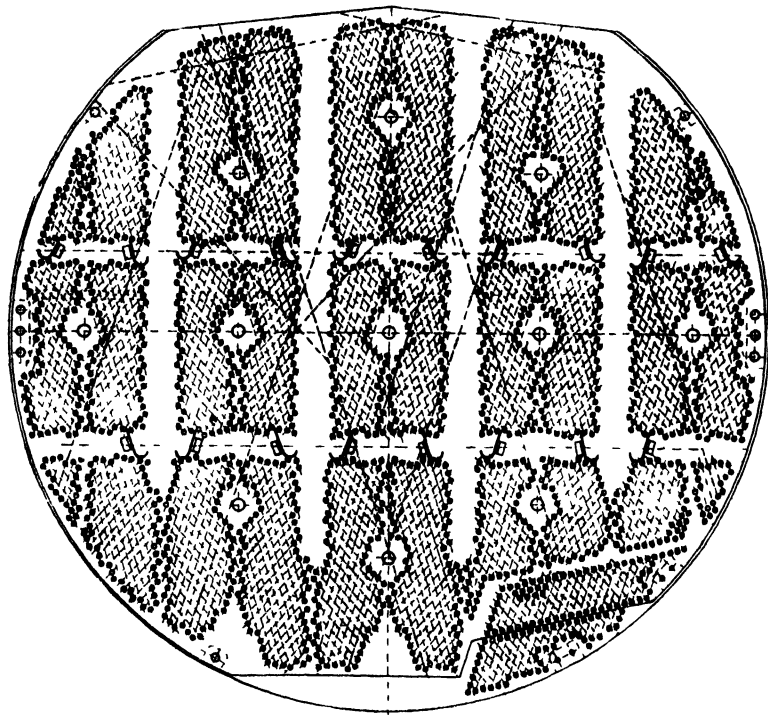


FIG. 50 (see p 106)

GINABAT CONDENSER BUILT BY SOCIÉTÉ DES CONDENSATEURS DELAS

The danger of blanketing of tubes and of the formation of air pockets is greatly reduced. Since the bulk of the steam condenses at the outer layers of tubes, it follows that the temperature of those closest to the gap is comparatively low and the temperature of the air is reduced as it passes downwards in contact with these cooler tubes.

Conversely, the condensate temperature is kept high since the greater part of the condensate only comes in contact with the hottest tubes.

Some interesting test results are quoted in Mr. Ginabat's paper, which show comparative readings between condensers built according

to older principles and according to those due to Mr. Ginabat, and in several cases it was found possible to carry out tests on two condensers which were duplicates of one another in all respects except in the matter of the tube layout and which were equipped with identical auxiliaries. It is, unfortunately, exceptional for designers to have the benefit of such an opportunity of checking the value of their improvements by means of strictly comparative tests carried out under working conditions on large units. The results in the present instance show a consistent advantage in favour of Mr. Ginabat's system. The case may be quoted of two condensers installed at the Comines Power Station of the Société de l'Énergies Electrique du Nord de la France. Each condenser has a cooling surface of 30,000 sq. ft., and has two water passes. Both condensers were cleaned out thoroughly a week before the test was carried out, and they were kept in commission for an equal length of time. The tests extended over a period of one hour, and the load on the turbines was maintained constant and at the same figure. In the case of the older condenser, the readings show $K = 274$, and in the case of that embodying Mr. Ginabat's designs, $K = 433$.

It is noted that these comparatively low figures are to be accounted for by the extensive fouling of the tubes which occurred since they were cleaned, but the advantage of the Ginabat design was sufficiently pronounced to induce the power company to change the design of the older condenser. The alteration could be effected with comparative ease, as the tube plates were the only parts to be replaced.

In the latest development of the Ginabat condenser by M. Delas, the tubes are arranged so as to give an increased area exposed to the direct flow of the steam. This is achieved by the bands of holes in the tube plates being in a saw tooth or accordion formation, and the bands being only about five tubes in depth. This gives a reduced pressure drop in the steam space, further increases the condensate temperature, and also allows the tubes to remain clean much longer than do the usual arrangements. Very extensive tests have been made at Gennevilliers, and the results that have been published are very satisfactory.

CHAPTER XI

CORROSION AND EROSION—SURFACE CONDENSER—FOULING OF CONDENSER TUBES

THE problem of condenser tube corrosion is one of the most difficult with which the modern power-house and marine engineers have been faced. It has engaged the attention of condenser makers and users and metallurgists in all industrial countries, and whilst the vast amount of money that has been expended on experimental and research work has thrown considerable light on the problem, it cannot be pretended that a complete solution of this problem has been reached. Remedies of troubles of one sort or another such as those referred to below have given most encouraging results in many cases, yet they have failed to be effective when applied to conditions that to all appearances presented an exactly similar problem.

Causes of Tube Corrosion. Tube corrosion may be caused by defective manufacture of the tubes. Simple chemical action due to impurities held in solution in the circulating water is a cause of corrosion which lends itself frequently to comparatively simple remedies. The practice of tinning the tubes or the kindred method of using Admiralty mixture tubes which contain 1 per cent of tin, have proved very effective safeguards. Many instances can be cited where costly corrosion difficulties have been entirely overcome by replacing tubes of the ordinary standard by tinned or Admiralty mixture tubes. For marine work and for land condensers using sea water, the use of such tubes has now become recognized standard practice. Experiments are now being carried out with tubes made of cupronickel, in some cases containing as high a proportion as 20 per cent of nickel, and the results obtained in various quarters are reported to be favourable. The use of chrome steel tubes has not apparently been tried so far.

Electrolytic Corrosion. One known cause of electrolytic corrosion is again defective manufacture, that is, the segregation of the constituents of the alloy which in conjunction with impurities of the circulating water set up an electric couple. A far more elusive cause of electrolytic corrosion is the presence within the water space of the tubes, of particles of free carbon. Since carbon is electro-negative to brass, the two materials form an electrolytic couple in the presence of sea water or any other acid or alkaline water, and the

electro positive element of the couple, in this instance the tube metal, dissolves, with the result that the tubes become pitted, and perforation ensues after a comparatively short time.

Carbon may be present in the circulating water in the form of ashes or soot, and its presence is difficult to avoid.

Cast iron exposed to sea water graphitizes the metal which flakes off, and this, in contact with the non-ferrous metals, sets up a very active electrolytic couple. Painting the surfaces of the castings exposed to the circulating water has frequently proved to be an effective remedy, but the paint usually wears away pretty rapidly and has to be renewed at frequent intervals.

Protection against Electrolytic Corrosion. Where it has not been possible to remove the cause of electrolytic corrosion, the practice of introducing renewable zinc blocks in the water heads is frequently adopted so that these may be attacked in preference to the tubes. This arrangement has failed to prove itself an effective remedy in many cases, as the zinc blocks polarize and eat themselves away without setting up an effective counter E.M.F. Another method of overcoming electrolytic corrosion is that of neutralizing the action set up by the electric couple within the condenser by impressing upon the system an opposite electromotive force. This principle is embodied in the Cumberland system.

THE "CUMBERLAND" PROTECTIVE SYSTEM. This system has been very extensively adopted in both land and marine work during the last few years, and is reported to have proved itself a very satisfactory means of overcoming corrosion of a severe nature.

In the case of the Brighton Municipal Electric Power Station, the system has been in successful operation for 12 years. Mr. John Christie, the Chief Engineer to the undertaking, reports that prior to its introduction the condenser tubes could not be relied upon for more than 6 to 9 months after completely re-tubing, whereas since the introduction of the system practically no trouble whatever has been experienced.

It would appear that continuous application of the system and careful attention to its proper working are essential to its satisfactory operation. Where this system is adopted it is necessary to ensure electrical contact between the tubes and ferrules in the manner already described.

The principle on which it is based is the introduction of insulated anodes into the water space of the condensers, consisting of cast iron blocks. From an outside source, by means of a motor-generator, these are charged positively with a current density of about $2\frac{1}{2}$ to 3 amperes per 1,000 sq. ft. of surface to be protected,

the flow of current being adjusted by suitable resistances, the E.M.F. behind the source being 6 to 10 volts, according to the conductivity of the electrolyte. The negative pole of the motor-generator is connected to the shell of the condenser, and the flow of this comparatively heavy current from the insulated anode deposited on the shell of the condenser, drowns out both poles of the little electrolytic couple, which works at milliamperes and millivolts, and which is the cause of the trouble.

The Coating of Tubes. Particulars have been received from Messrs. Charles Howson & Co., Ltd., of Liverpool, of a plant patented by Mr. J. Austin, and designed to safeguard condenser tubes against corrosion by covering the surface with a coating of bitumastic solution. The inventor has published particulars of his apparatus in a paper read before the Liverpool Engineering Society.* It would appear that the apparatus consists of a spraying plant designed to ensure that the coating of solution can be evenly applied to the tubes of a condenser in place, and the results recorded so far indicate a substantial improvement of the life of the tubes treated in this manner. Particulars are not available of the effect of the coating on the heat transmission coefficient, but since the layer of solution is thin the added resistance is not likely materially to reduce the value of K .

Causes of Tube Erosion. The causes of tube erosion on the other hand are less complicated in their nature. Defective production of the tubes, and the presence of foreign matter in the circulating water, are causes that are comparatively easily located.

The Fouling of Tubes. The effect of variations of the heat transmission coefficient on the vacuum that can be obtained in a given condenser has been fully considered in Chapter V. It has also been shown that in calculating the surface area an allowance should be made for a margin on account of fouling of condenser tubes. Where the essential causes, whatever these may be, for fouling of tubes persist for prolonged periods, any available margin will naturally disappear, and the vacuum will fall below economical limits and necessitate shutting down for cleaning purposes. The fouling of tubes may be due to a variety of causes, for instance, the presence of foreign bodies such as leaves and other vegetable matter which are often present for certain periods of the year only, and which, incidentally, can generally be cleared out of a condenser in a comparatively short time, or again, we may have to deal with hard-setting scale which is very difficult to remove.

* J. Austin: "Methods of Preventing Corrosion of Condenser Tubes," Liverpool Eng. Soc. 51st session.

FORMATION OF SCALE IN THE TUBES. The formation of scale in condenser tubes is generally primarily caused by the presence of calcium carbonate or carbonate of lime amongst other constituents in the circulating water. Carbonate of lime is not soluble in pure water, and its presence in solution in water is generally due to the fact that the water contains carbon dioxide with which the carbonate of lime combines to form calcium bicarbonate. This combination is in a very unstable condition, and a change in temperature of the

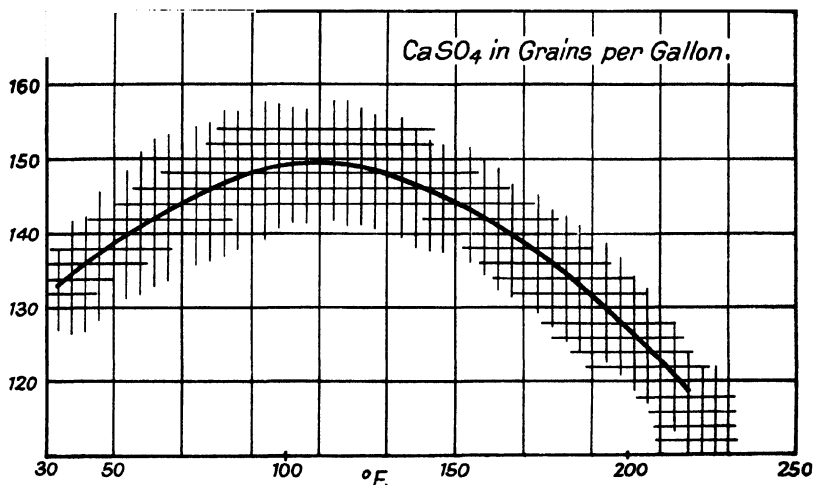


FIG. 51
SULPHATE OF LIME CONTENTS IN WATER.

water or of its pressure easily leads to the setting free of carbon dioxide, leaving particles of carbonate of lime in suspension. Carbonate of lime when deposited forms a soft sludge rather than a hard scale, but once such a deposit starts to form, sulphates of lime and of magnesia are readily brought out of solution and combine with the carbonates to form a hard scale which it is very difficult to remove. Although far more carbonate of lime than sulphate of lime is actually deposited, the analysis of a condenser scale given below does not show the difference between the quantities of these two constituents one would expect. This is, no doubt, to be accounted for by the fact that the greater proportion of the softer carbonates is carried away by the water before it has time to combine with the sulphates and to form a hard scale.

The curve shown in Fig. 51 has been plotted from figures quoted

by Mr. J. H. Paul,* and shows the amount of sulphate of lime which water is capable of containing at different temperatures. From this curve it will be seen that if water entered a condenser at a temperature of 60° F., carrying the maximum of 141·5 grains of sulphate of lime per gallon in solution, then the water temperature would have to rise to 158° F. before it becomes supersaturated, and consequently, if it were not for the action of the free carbonate of lime in bringing the sulphate out of solution, this would not be likely to be deposited at all in a high vacuum condenser.

RATE OF FORMATION OF SCALE. If the whole body of the water were passing through the condenser at the mean water velocity of 3 to 7 ft., there would hardly be sufficient time for the change to take place and for the suspended carbonate of lime to be deposited. It has been shown that whatever the mean water speed there is always a viscous film of water at the outer periphery of the plug of water passing through a tube, whose velocity varies from zero at the outer surface in contact with the tube to a maximum at the stage where the viscous film joins the turbulent core. It seems likely to be due to the presence of this viscous film that scale is formed, and its rate of formation is most probably a function of the thickness of the film and of the temperature of the tubes and of the temperature and velocity gradients across the film. The very fact that scale is formed in high vacuum condensers where the circulating water is hard, practically furnishes a proof of the existence of the viscous film where the bulk of the water is turbulent. For not only is the water velocity of the turbulent core probably too great to allow a deposit to be formed, but the temperature rise of the core in the immediate vicinity of the water inlet is too slight to admit of any substantial amount of scale-forming deposits being released. In practice it is, however, found that the thickness of scale formed is fairly uniform throughout a condenser.

THE SCALE-FORMING PROCESS. The formation of scale has been observed to start at the top tubes at the water outlet end. One can readily picture the process of a gradual formation of a scale of uniform thickness in the following manner. Where a condenser is clean the most favourable conditions for the formation of scale prevail in the top rows of tubes at the water outlet end, for at this stage the tube temperature is a maximum, and the temperature gradient across the viscous film is a minimum and the water pressure is a minimum. As soon as a layer of scale is deposited in the

* *Boiler Chemistry and Feed Water Supplies*, by J. H. Paul, B.Sc., F.I.C. (Longmans, Green & Co., 1919.)

outlet end of a tube, then the interposing of this scale, with its high resistance to heat transfer, substantially increases the temperature gradient between the steam and the outer film of circulating water, so that the temperature of the outer layer and of the layers of water immediately adjoining this are lower than in portions of the tube which are still comparatively free from scale. Consequently, the rate of formation of scale becomes lower than in the adjoining part of the tube, so that the point at which scale is deposited at its maximum rate moves away from the water outlet end of the tubes, and this process continues until the whole tube system is fairly evenly coated with scale.

If the scale is set hard, its removal presents considerable difficulty. It has either to be removed mechanically, usually by drilling, or it can be dissolved by boiling out the condenser with diluted hydrochloric acid. Both processes are cumbersome and costly.

Effect of Scale on Efficiency of Condenser. The figures given in Table VIII represent a series of readings taken on Condensing Plants, in use at the Commercial Road Electric Power Station of the Wolverhampton Corporation, and kindly supplied to the authors by Mr. S. T. Allen, Chief Engineer of the Electricity Department. These readings were taken under ordinary commercial running conditions, the instruments used being the station instruments which had not been specially calibrated for the purpose. Consequently they can only serve the purpose of providing approximate values.

TABLE VIII
READINGS SHOWING EFFECT OF FORMATION OF SCALE ON CONDENSERS
AT WOLVERHAMPTON CORPORATION ELECTRIC POWER STATION.

	(1)	(2)	(3)	(4)
Date	9/2/23	5/3/23	16/6/23	26/7/23
Lbs. of steam per hour	61,000	62,000	53,000	59,000
Bar	28.92"	29.33"	29.36"	29.37"
<i>Vacuum—</i>				
Gauge	23.2"	27.8"	24.7"	27
Corr. to 30 in bar	24.05	28.45	25.28	27.62
t_1 ° F.	59	62	78	70
t_2 ° F.	82	69	94	84
<i>Scale—</i>				
Top	0.088"	nil	0.053"	nil
Bottom	0.088"	nil	0.053"	nil
K	77.5	204	106.5	219
K scale	10.9		15.6	
Effective tube area, sq. in	0.413	0.638	0.235	0.336
Relative water velocity	0.46	1	1.1	1

ANALYSIS OF SCALE

Lime, CaO	30.20%
Magnesia, MgO	18.30%
Chlorim, Cl.	5%
Carbon dioxide, CO ₂	12.00%
Sulphate radicle, SO ₃	30.20%
Combined water, H ₂ O	4.30%

NOTES. Readings (1) and (2) refer to a two-pass condenser having a surface area of 11,400 sq. ft., tubes 1 in. diameter, No. 18 S.W.G. Distance between tube plates 16 ft.

Readings (3) and (4) apply to a two-pass condenser having a surface area of 9,400 sq. ft., tubes $\frac{3}{4}$ in. diameter, No. 18 S.W.G. Distance between tube plates 13 ft.

$$K_{\text{scale}} \text{ calculated from } \frac{1}{K_{\text{scale (1)}}} = \frac{1}{K_{(1)}} - \frac{1}{K_{(2)}}$$

$$\text{and } \frac{1}{K_{\text{scale (3)}}} = \frac{1}{K_{(3)}} - \frac{1}{K_{(4)}}$$

and brought to a common thickness of 1 in.

The relative water velocity has been calculated on the basis of the ratio of water quantity determined from the water temperature readings to effective tube area.

The method of calculating K_{scale} does not allow for variations in the heat transmission coefficient, due to causes other than the formation of scale. Thus, in the case of readings (1) the water velocity was perceptibly lower than for series (2), so that the heat transmission coefficient K would be lower in the former case and not the same as assumed, and K_{scale} consequently higher than the calculated value, but the available data are not sufficient to make the necessary corrections. It should also be noted that the air quantity entrained and the air pump capacity may have varied between readings.

Cleaning without Interrupting Service. Condenser makers have devoted a great deal of attention to the problem of building condensers which can be cleaned without interrupting their service. Fig. 52 shows an arrangement of condensing plant built by Messrs. Richardsons-Westgarth, about 20 years ago, consisting of two complete units with sluice valves in the exhaust connections which can be operated to keep one plant in commission whilst the other one is being cleaned out. This early attempt to deal with the problem is naturally expensive in first cost and cumbersome, so much so as to rule it out as a practicable design where modern large turbine units, having exhaust branch areas running into hundreds of square feet, are involved.

Brown Boveri Continuous Service Condenser. Messrs. Brown Boveri & Co. appear to have been the originators of the so-called Continuous Service Condenser. This consists in principle of two complete condensing plants, the two condenser bodies being built into a single shell and having a common steam space, but having separate waterways and preferably hinged doors. Under normal working conditions both halves of the plant are kept in operation, and the plant as a whole is usually designed to maintain the specified

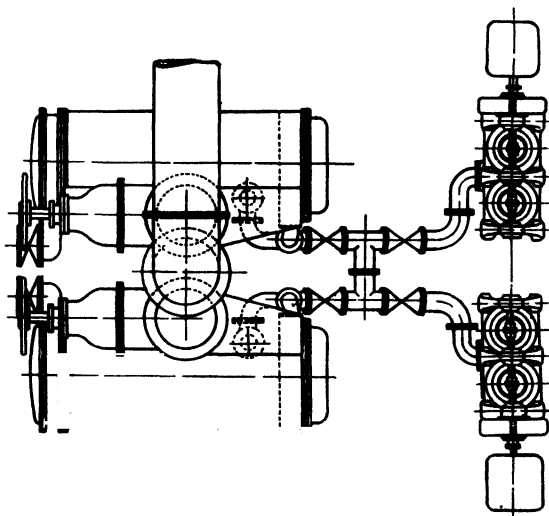


FIG. 52.

ARRANGEMENT OF DUPLICATE CONDENSERS AND AIR PUMPS.

vacuum when working in this manner. When it becomes necessary to clean the condenser, then it is only necessary to shut down one circulating pump and its valves and drain the water from this half. Both air and both condensate pumps can be kept running if it is found convenient to do so. The condenser doors on one section can then be opened, and one half of the condenser can be cleaned out without interfering with the service of the other half, since the fact that a vacuum is being maintained on the steam side of the section being cleaned does not matter in any way, except that it is not possible to replace tubes or tube packings under these conditions.

ADVANTAGES OF CONTINUOUS SERVICE. The advantage of the arrangement arises from the fact that practically all modern condensers built for turbine work are designed on the flat part of a

curve, whose ordinates are vacuum and whose abscissae are water quantity or surface. In other words, the reduction of the available cooling surface and cooling water quantity to half their normal values only involves a comparatively slight fall in vacuum, which the user can afford to sacrifice during the period of carrying out cleaning operations far more readily than he could face putting the complete plant out of operation. Whilst the necessary use of

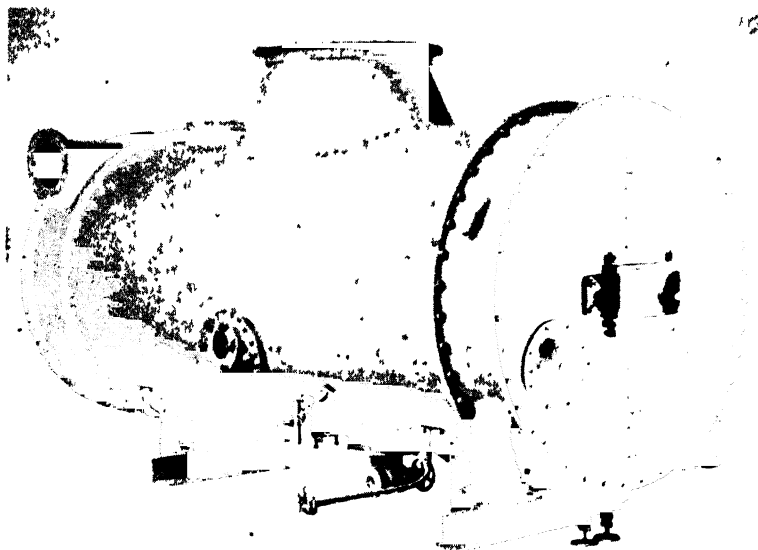


FIG. 53.

BROWN BOVERI CONTINUOUS SERVICE CONDENSER, SHOWING
HINGED DOORS

duplicate auxiliaries of half capacity involves an additional capital expense and a slight additional expenditure in power consumption, this drawback is neutralized by the advantage gained by the reduced liability to a complete loss of vacuum in the event of a single pump failing.

Under certain conditions an additional advantage arises from the use of such an installation in that one half of the plant can be shut down on light loads. This applies particularly in the case of pass-out turbine installations, where the low-pressure steam demand is limited to certain parts of the day or of the year, as, for instance, in a case where the turbine runs fully condensing in summer time but where the bulk of the exhaust steam is required for heating

purposes during the winter. In such a case, a marked saving is effected by running only one set of auxiliaries during the winter months.

Figs. 53 and 54 give a photographic and a sectional illustration of Messrs. Brown Boveri's Continuous Service Condensers. The former shows the water doors in two parts, and one-half can be swung clear of the condenser without disturbing the other half. The O.V. design, already referred to, enables the makers to adapt their standard design to the continuous service design, with the simple addition of vertical division plates in the waterways, and of double inlet and outlet branches as shown.

Fig. 55 shows a 7,000 sq. ft. condenser built on similar principles, due to The Worthington Pump & Machinery Corporation, of New York. In this case, a somewhat different method of hinging the doors has been adopted.

Fig. 56 shows a continuous service condenser built by The Ingersoll Rand Co. In this case, the division line is on a horizontal plane. It will be seen that the inspection doors are hinged, but the doors themselves are not hinged.

The Hulsmeyer Self-cleaning Condenser. Messrs. Maschinenfabrik Augsburg Nürnberg (M.A.N.) have developed a condenser design, known as the Hulsmeyer type, in which it is claimed that the fouling of the condenser tubes is avoided by temporarily increasing the water velocity in one of a number of compartments at a time. It is claimed that the water velocity is raised to such an extent as to dislodge any foreign matter or scale that may be in process of formation, and that this can be effectively carried out without interrupting the service of the condenser in any way, once every day or every week, according to the nature of the water. The same object could, of course, be attained by temporarily raising the circulating water speed and thereby increasing the water velocity

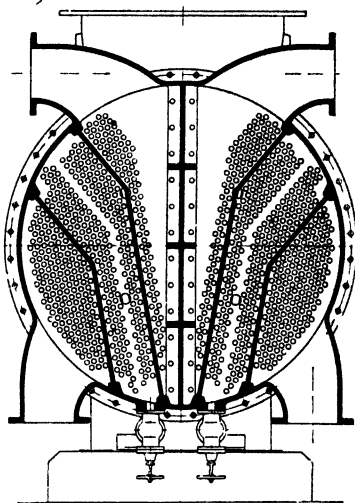


FIG. 54.
SECTIONAL VIEW OF BROWN BOVERI
CONTINUOUS SERVICE CONDENSER.

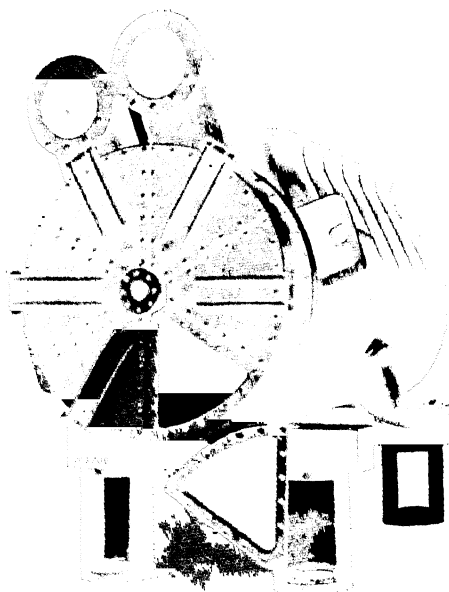


FIG. 55 (see p 119).
CONTINUOUS SERVICE CONDENSER BY WORTHINGTON
PUMP AND MACHINERY CORPORATION

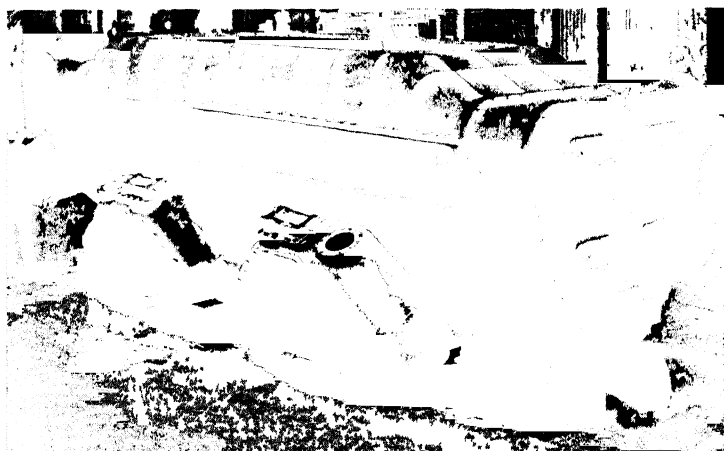


FIG 56 (see p 119)
INGERSOLL RAND CONTINUOUS SERVICE CONDENSER.

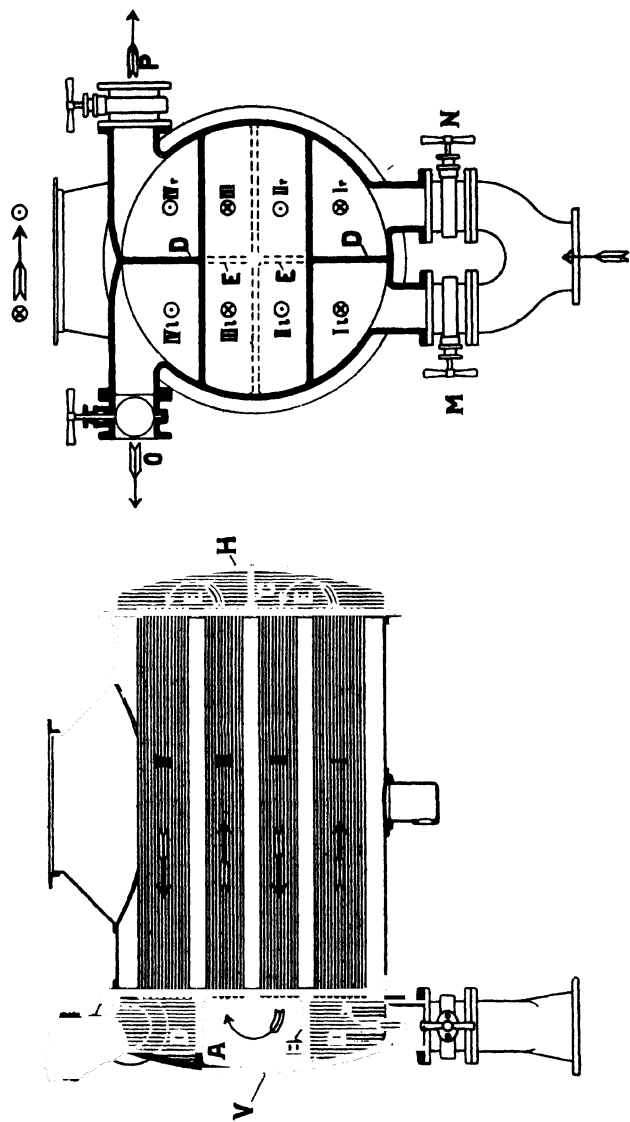


FIG. 57 (sec p 119)
HULSMAYER CONDENSER (M A N)

throughout the condenser, but the additional friction head, coupled with the increased water quantity, would call for abnormal loading of the pump and its driving motor or turbine, with the result that the normal running conditions would necessarily be very inefficient.

By confining the raising of the water velocity to two compartments at a time, this difficulty can be avoided. Fig. 57 gives a diagrammatic view of such a condenser. The condenser is of the four-pass type, and is provided with two inlet branches M and N, and two outlet branches O and P, and each branch is fitted with a throttle valve. The front water chamber V is provided with two horizontal division plates, A and B, and the upper and lower chambers are provided with vertical division plates D. The back water chamber H has a single horizontal division plate C, and a vertical division plate E. Under normal working conditions all four throttle valves are kept fully open, and the water passes through in two parallel equal streams through the various passes at constant velocity in the manner indicated in the illustration. If it is intended to flush compartments I_r and II_r, then it is only necessary to close valve M. The whole of the water will then enter the condenser through valve N and pass with, approximately, twice normal velocity through sections I_r and II_r, the water remaining stationary in I_l and II_l, and passing at normal velocity through the remaining sections. To flush sections I_l and II_l, valve M is left open and N has to be closed. Similarly, to flush sections III_r and IV_r, the outlet valve O should be closed, and, lastly, to flush sections III_l and IV_l, valve P has to be closed.

It is claimed that each flushing operation can be carried out effectively in 10 to 15 minutes, so that the whole condenser can be flushed within an hour. Since the increased friction head due to raising the water velocity is only that due to a quarter of the total body of tubes, it has been found possible to attain an increase of 75 per cent in water velocity for flushing purposes with a pump of standard design. The whole cleaning operation can be carried out under any conditions of load, as the loss of vacuum is very slight, but it is considered most advisable to choose a condition of light load. The makers suggest the use of butterfly type valves for convenience of handling.

Mirrlees-Watson Vertical Condensers. A case of exceptional difficulty from the circulating water standpoint is that of the Leeds Corporation Electric Power Plant, where the circulating water is drawn from the River Aire. During certain times of the year, the river water is greatly polluted by industrial refuse of a fibrous nature, such as woollen waste and the like, and the difficulties

of clearing such materials are self-evident. Messrs. Mirrlees-Watson originated a special vertical condenser designed to cope with these conditions. The two first condensers of the kind

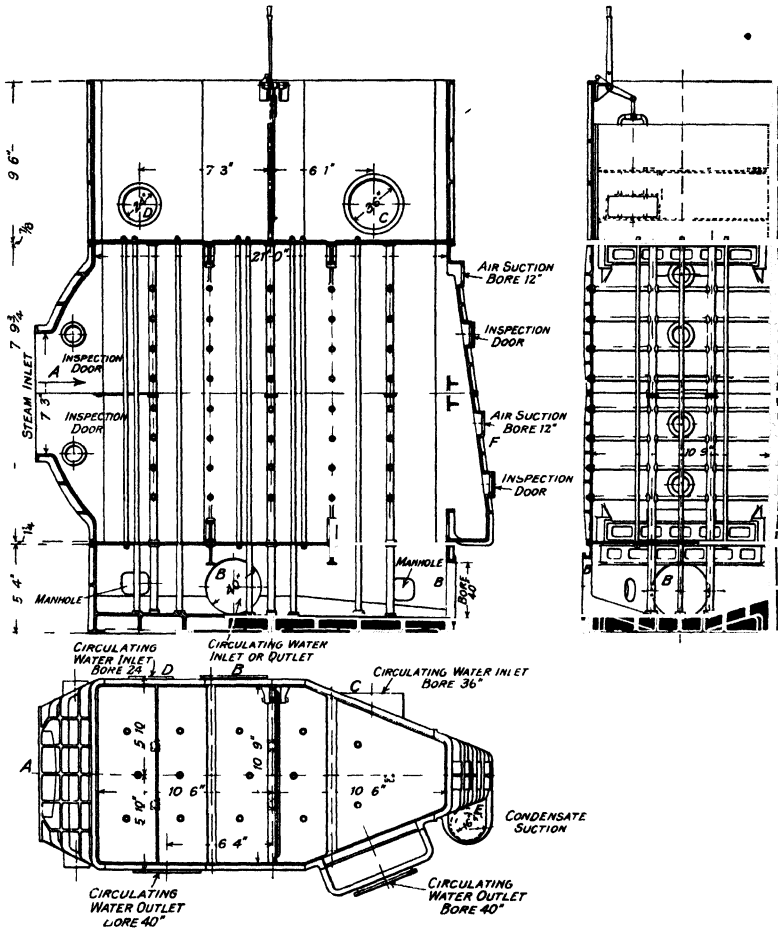


FIG 58

LEEDS VERTICAL CONDENSER BY MIRRLEES-WATSON LTD

were installed in 1906, and were connected to Willans 3,000 kw. turbines. These were followed by two units coupled to 6,000 kw. turbines, and a later condenser is connected to a Zoelly 12,000 kw. turbine, built by Messrs. Escher Wyss. In these condensers the upper water box is in the nature of an open tank, so that the

individual tubes can be cleaned from the platform at any time whilst the plant is in operation. The steam passes from the horizontal exhaust flange of the turbine through a pipe forming a right-angle bend to the steam inlet. This pipe has a bore of 7 ft. 3 ins. The condenser is built up of cast iron sections, and has an over-all height of 37 ft., its total weight being 135 tons.

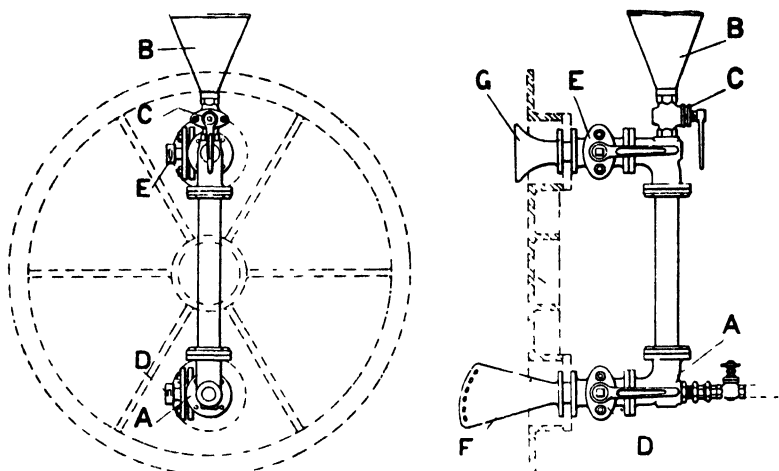


FIG 59

MIRRELES-WATSON TUBE CLEANING DEVICE.

The tubes are exceptionally large, namely $1\frac{1}{8}$ ins. external diameter, 18 S.W.G. thick, and the length between tube plates is 24 ft.

A false tube plate is fitted with trumpet-shaped openings to avoid sharp edges which might tend to catch foreign matter. The method of supporting the tube plate and other details of construction are shown in Fig. 58.

The condenser works normally with a single water pass, the water entering at the top and leaving at the bottom, but when the water is clear it can be arranged to pass twice. To enable this to be accomplished, a diaphragm plate is fitted in the upper water head or tank and can be lowered or raised as required. Provision is also made for reversing the direction of water flow for the purpose of washing out.

The condenser has a surface of 46,000 sq. ft., and the plant maintains a vacuum of $28\frac{1}{2}$ ins. when dealing with 220,000 lbs. of steam per hour, and when supplied with 36,000 gals. of water per minute at 65° F. This performance is obtained when the condenser

is working under single flow conditions when the water velocity is 1.9 ft. per second. The heat transmission coefficient is naturally low.

The air pump is of the Mirrlees-Leblanc type, and this, together with the Hotwell pump and a boiler feed pump, are driven in a line by an auxiliary turbine.

The Mirrlees-Watson Tube Cleaner. The tube cleaning device illustrated in Fig. 59 is due to Messrs. The Mirrlees-Watson Co., Ltd., of Glasgow, and is designed to clean condenser tubes by chemical process. It consists of a steam ejector located in a suitable piping system, and is so arranged that a cleaning liquid can be easily introduced and controlled. The action of the ejector causes the solution to circulate in the condenser and also to be heated to the required temperature. The nature and quantity of the cleaning liquid to be used, and the length of time necessary for the solution to be circulated in the condenser, depend on the character and amount of deposit to be removed. Where this is of a greasy or muddy nature, the makers recommend the use of caustic soda, using first a 1 per cent solution and increasing the strength up to 7 per cent as a maximum. Where hard scale has to be treated, diluted hydrochloric acid is recommended, starting with a 5 per cent solution. If this is ineffective, then the strength should be increased to a maximum of 40 per cent, but this limit should not be exceeded.

The method of operation is as follows. Having closed the circulating water valves and opened the air cocks on the tops of the water heads, almost fill the condenser with a known quantity of clean water; close the air cocks and then open the valves E and D, referring to Fig. 59. If steam is now admitted to the ejector A, the water in the upper part of the condenser is drawn through the bell-mouthed orifice G into the cleaning apparatus and, passing down the ejector, is discharged through the perforated cone F into the lower part of the condenser, the water at the same time being heated by the steam from the ejector.

When circulation is thus established, a quantity of the cleaning liquid sufficient to form the required solution is poured into the funnel B, and, on opening cock C, the cleaning liquid is drawn in and mixed with the hot water circulating in the condenser.

The cleaning operation should be continued for at least 6 to 8 hours, and as the solution gradually becomes weaker, the required strength must be maintained by adding cleaning liquid from time to time. When a dirty condenser has been restored to its initial condition of cleanliness frequent periodical cleaning should be

maintained, and then each cleaning operation would occupy but an hour or so. The strength of the cleaning solutions must not be allowed to exceed the limits prescribed, and the temperature must be kept within a maximum of 150° F. by occasionally shutting down the ejector.

When the cleaning operation has been concluded and the liquid drained off or pumped out, the condenser must be washed out thoroughly with clean water when caustic soda has been used as the cleanser ; when hydrochloric acid has been used, the condenser must be washed out several times with an alkaline solution. Any tendency for the action of the acid to continue after washing out is soon neutralized by the chemical reaction with the scale-producing impurities in the circulating water.

CHAPTER XII

EVAPORATIVE CONDENSERS

IN certain situations it is difficult to justify the expense of installing a comparatively expensive surface condensing plant, and recourse may then be made to evaporative condensers.

In these, a large surface is provided and exposed to the atmosphere, water being circulated over the surface.

The heat of the steam condensed is dissipated by two means—

(1) Direct heating of the air which comes in contact with the surface, and

(2) Evaporation of a portion of the water which passes over the surface.

Usually both means are utilized at the same time but, as will be shown later, it is possible at low loads to rely solely upon the first method.

The Ledward & Beckett Condenser. Before dealing with the theory and with certain special points, a representative evaporative plant will be described. In Fig. 60 there are shown two views of a plant made by Messrs. Ledward & Beckett. Steam enters the condenser at the top after passing through a grease separator (these condensers are used more frequently with engines than with turbines), and is conveyed from the main header into a number of banks of gilled pipes in which it is condensed. The resulting condensate and the air are drawn away by an air pump, usually placed directly under the condenser. A centrifugal pump is shown, belt driven from the air pump, and this circulates the cooling water from the tank under the condenser to the water distributing pipes over each bank of gilled pipes.

The gilled pipes are shown in greater detail in Fig. 61. At the top there will be seen the water distribution pipe with holes in the top. Water flows over the surface of this pipe and drops on to the gilled pipe from the serrated bottom edge, as shown in the side elevation. If the water pressure is sufficient, the water rises to and falls back from the deflecting cap on to the distributing pipe. The section adopted for the gills on the pipes is the result of much experimental work, the object of which was to obtain a form which would retain the water in contact with the tube in a wind. The section shown is found to be very effective.

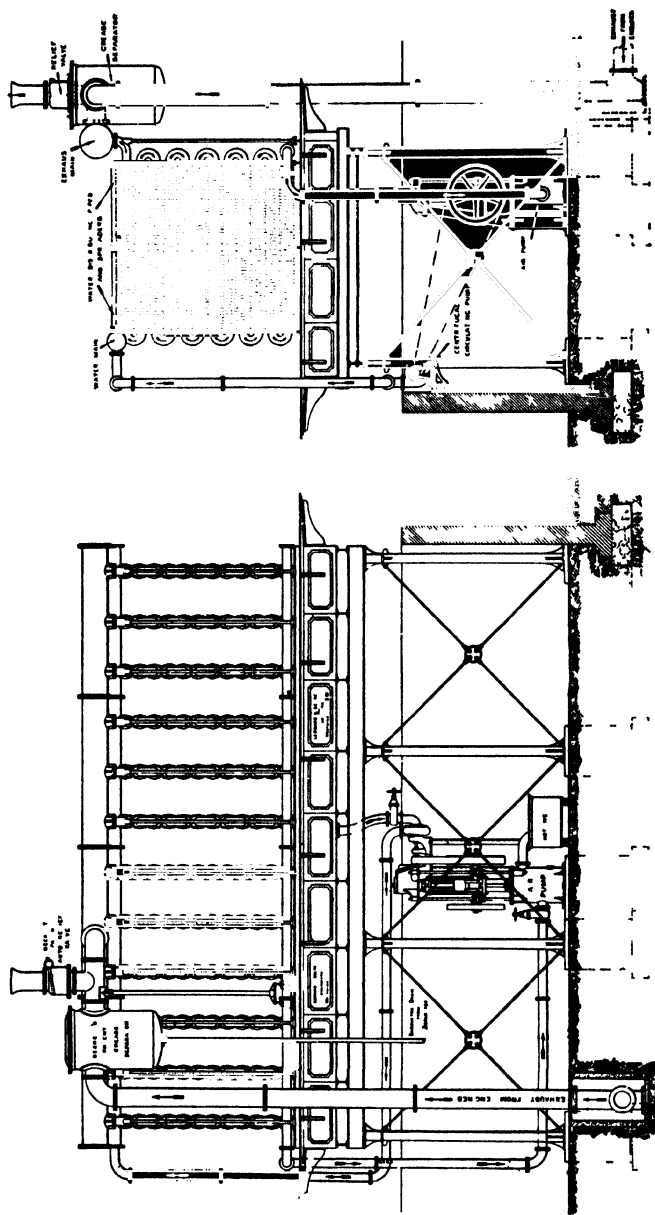


FIG. 60 (see p. 127)
EVAPORATIVE CONDENSER ARRANGEMENT
(Ledward & Beckett, Ltd.)

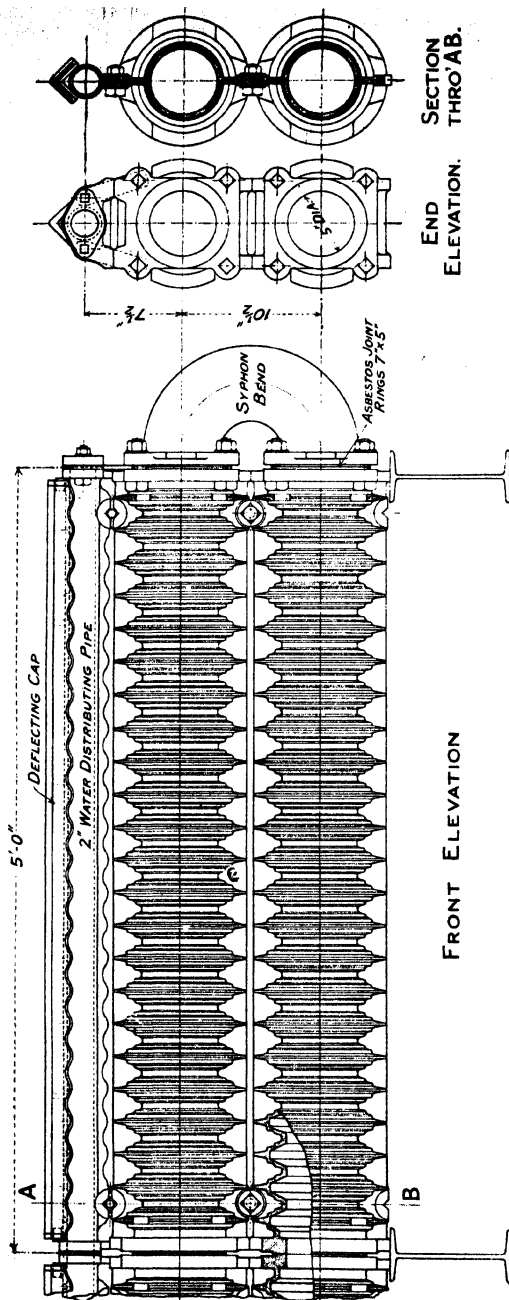


FIG. 61 (see p. 127).
GILLED TUBES FOR LEDWARD & BECKETT EVAPORATIVE CONDENSER.

Relative Advantages of Both Types of Condensers. An evaporative condenser combines the functions of a surface condenser and an open type water cooler. Thus, in a surface condenser, the steam heat is transferred to the cooling water, and the heated water is then passed through a cooler. In the cooler it is made, in falling, to present a large surface to the ascending current of air and so passes on its heat to the air. In the cooler, as in the evaporative condenser, heat is transferred both by direct heating and by evaporation of a portion of the water.

In the evaporative condenser the steam is taken to a structure which may be compared to the open type cooler, except that the surface is metallic instead of aqueous, and the heat transfer takes place directly between the steam and the atmosphere through the surface.

In a cooler, or in a cooling tower, roughly about 70 per cent of the heat is removed by actual evaporation of a portion of the water, and 30 per cent by heating of the air. In an evaporative condenser, the same ratios are about correct when the plant is working under full load. For light loads the condenser has one great advantage, in that the direct air heating practically remains constant, and the amount of water evaporated is reduced with the load. In actual practice, it is found that the water circulation can be stopped entirely when the load on the plant is about 30 per cent of full load, or even more. In some cases, the water can be stopped at half load, the exact ratio depending upon the proportions of the plant and on the prevailing atmospheric conditions.

In a surface condensing plant the same quantity of water is usually circulated at all loads, so that the power used for this purpose becomes of relatively greater importance at low loads. Not only is it possible to shut down the circulating pump at low loads on an evaporative condenser, but the circulating water power consumption on full load is very much less with this type of condenser than with a surface plant, as the ratio of water to steam is much less and the head on the pump is also less.

In a water cooling tower the water lift is not less than 18 ft., and in some cases as much as 24 ft. In the condenser shown in Fig. 60 the actual water lift is 12 ft. 6 ins., this being the height of the water distribution pipe above the water level in the tank.

ECONOMY IN AUXILIARY POWER DUE TO EVAPORATIVE CONDENSER. Thus in a surface condenser there may be circulated 50 lbs. of water per lb. of steam (and this is considered to be a small amount) against a head of 20 ft., whereas in an evaporative condenser the quantity circulated will only be about 10 times and the head 12 ft.

6 ins., so that the power used on full load will be $\frac{50 \times 20}{10 \times 12.5}$ or 8 times as great with the surface plant. The air pump power is constant in both cases and runs continuously, so that the total power ratio is not as high as 8 to 1.

There is a plant of this type at Paignton, Devonshire, designed to deal with 8,000 lbs. of steam per hour and to maintain a vacuum of 26 ins. This plant replaced an older plant using a cooling tower, and the following figures show conclusively the saving in auxiliary power—

	1921. Old Condenser.	1922. Evaporative Condenser.
Units generated	534,440	578,743
Units used on condenser	78,144	19,079
	14.6%	3.3%

The drawbacks of this type of plant are mainly the space occupied, and the fact that a high vacuum, judged by turbine standards, cannot conveniently be obtained. As to the space occupied this drawback can be minimized by placing the condenser upon a roof or on a steel structure in the yard. Elevating the condenser in this way is beneficial, as it places it in a more exposed position with better cooling effects.

There is a considerable vacuum drop between the engine and the air pump suction. Thus, in a large plant at Southend, the vacuum at the air pump suction was 26 ins., and at the engine 25 ins. The exhaust main between the pump and engine in this case is 130 ft. long.

The Southend plant was designed to maintain a vacuum of 26 ins. when dealing with 60,000 lbs. of steam per hour. The total surface is 85,500 sq. ft., divided between 99 coils or banks of tubes.

This plant is installed in an overhead tank, 96 ft. long by 48 ft. wide by 24 ins. deep. It is provided with two three-throw pumps of the Edwards type, with barrels 12 ins. diameter by 10 ins. stroke, running at 118 revolutions per minute. This pump equipment corresponds to a displacement of about 0.46 cu. ft. per lb. of steam. Ten lbs. of water are circulated per lb. of steam. Lengthy experience has shown that, on full load, about two-thirds of a lb. of water are evaporated per lb. of steam condensed. As the latent heat at

26 ins. vacuum does not differ greatly from that at atmospheric pressure (at any rate for a calculation of this nature), this means that 67 per cent of the heat was removed by evaporation and 33 per cent by direct air heating.

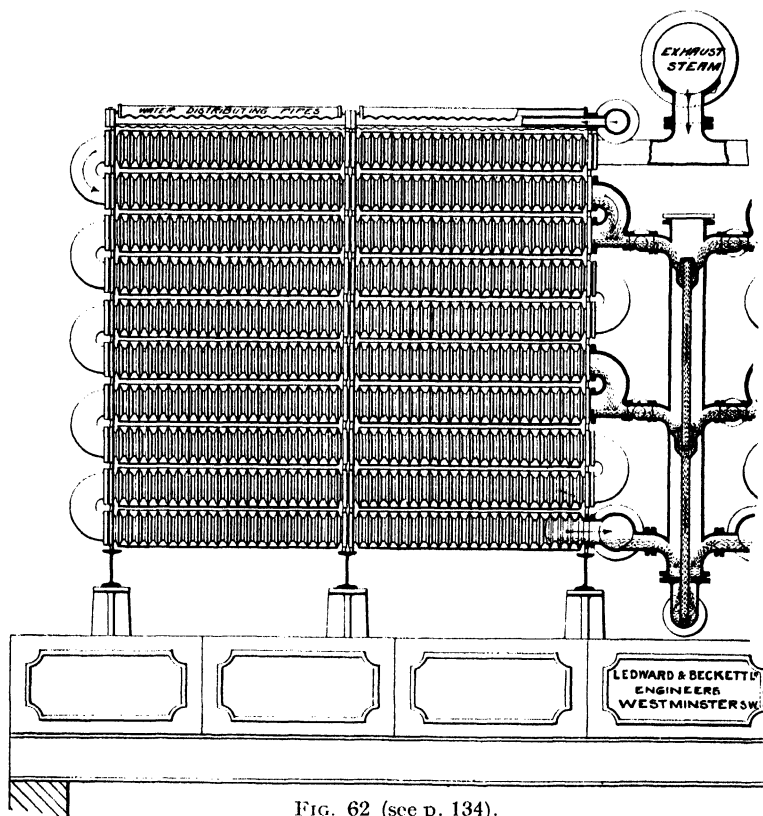


FIG. 62 (see p. 134).

LEDWARD & BECKETT EVAPORATIVE CONDENSER, SHOWING
COMPARTMENTAL DRAINAGE.

The Use of Evaporative Condensers. The amount of water evaporated in one of these condensers is about the same as in a natural draught cooling tower, so that no claim can be made that it reduces the amount of water that has to be purchased for make-up purposes when the plant is on full load. When, as is usually the case, the plant is only fully loaded for an hour or two in the twenty-four, the evaporative condenser shows to advantage,

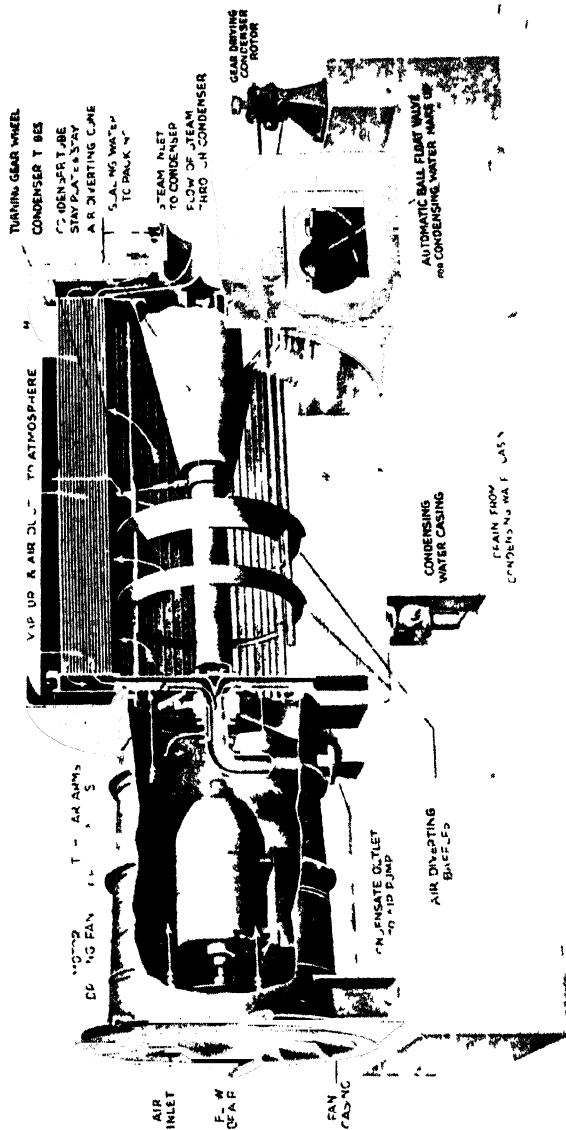


FIG 63 (see p 134)
 DIAGRAMMATIC SECTION OF RAMSEY EVAPORATIVE CONDENSER

due to the fact that the water can be stopped and evaporation losses avoided except at higher loads. This coupled with the saving of power, as evidenced by the Paignton figures, gives the evaporative condenser considerable advantages in a power station having a small day load, particularly if the cost of water is high.

There is another point about evaporative condensers which may be either a disadvantage or an advantage, according to the circumstances. The volume of the condenser with the exhaust main leading from the engines is so large when compared with the volume of the usual surface or low level jet plant that, if the Edwards pump is the only air extractor, it takes a relatively long time to establish the vacuum. This is a drawback, but should anything cause the air pump to stop, the large evacuated volume becomes an advantage as the vacuum does not drop as rapidly as in a more compact plant.

The plant may be arranged so that the condensate from each pair of coils in a vertical bank of tubes is led direct to a central stand pipe, as shown in Fig. 62. In this stand pipe it passes through a water seal to the air pump suction, or to the extraction pump if fitted separately from the air pump.

The "Ramsey" Condenser. A more compact form of surface evaporative condenser has been introduced by Ramsey, and has been made by Fullerton, Hodgart & Barclay, of Paisley. A diagrammatic section is shown in Fig. 63. It has been used for small-power plants in mills, but the most interesting installation is in a locomotive built for the Ramsey Condensing Locomotive Co., Ltd., by Messrs. Sir W. G. Armstrong, Whitworth & Co., Ltd. In Fig. 64 there is shown a photograph of the locomotive condenser. From these two figures the action will be clear. The annular nest of tubes rotates slowly, water being kept at a certain level by the ball float shown and submerging the lower tubes. Steam enters the condenser by the end flange shown on the right of Fig. 64. At the other end there will be seen some of the blades of a motor-driven fan, the inlet to which is placed at the front of the locomotive. Air is thus forced over the wet tubes and condenses the steam by removing the heat mainly by evaporation of water as in the larger type of evaporative condenser.

This particular locomotive condenser has been tested and, by the courtesy of Mr. G. F. Jones, the following particulars are available:

The effective cooling surface was 3,050 sq. ft., and the steam condensed 15,000 lbs. per hour. The operating water was at a temperature not exceeding 120° Fahr. approximately. The plant was fitted with a two-stage steam air ejector by the Mirrlees-Watson

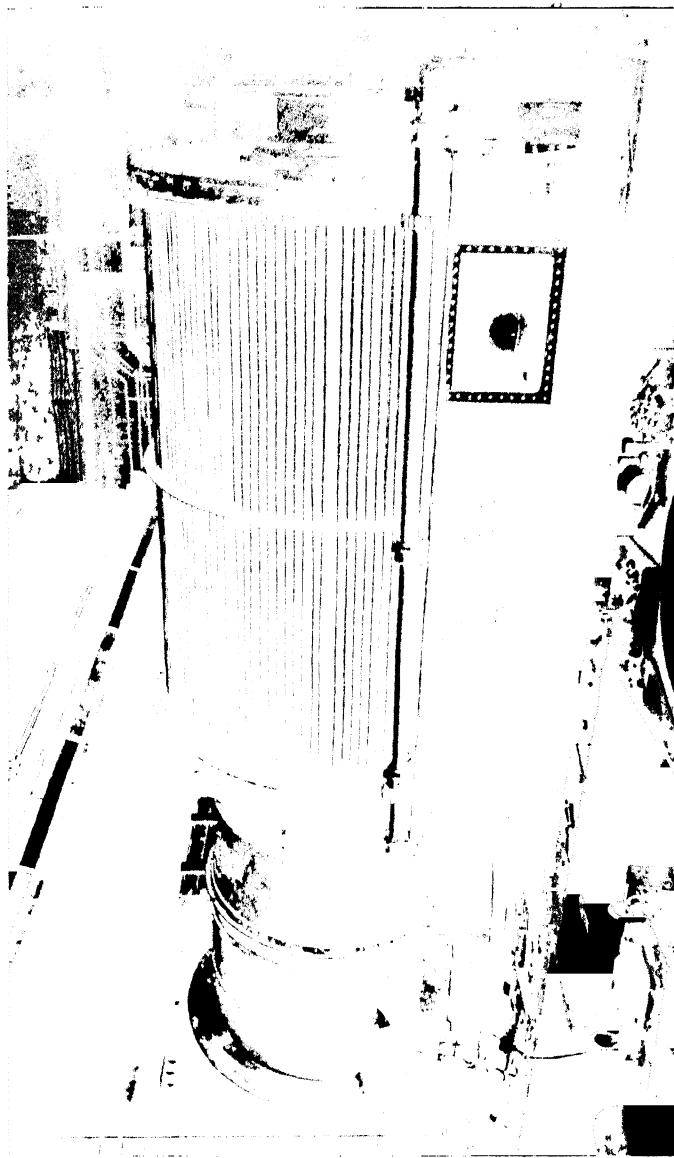


FIG 64
LOCOMOTIVE CONDENSER

Co., Ltd., and the average vacuum was 26 in. (Bar. 30 in.). As the fan was driven by a direct-current motor, the quantity of air blown through the condenser could be varied with the temperature. The relation between power absorbed by the fan and the temperature of the inlet air is shown in Fig. 65.

The experience gained with this condenser has led the inventors to an improved design of condenser, as shown in Fig. 66. The

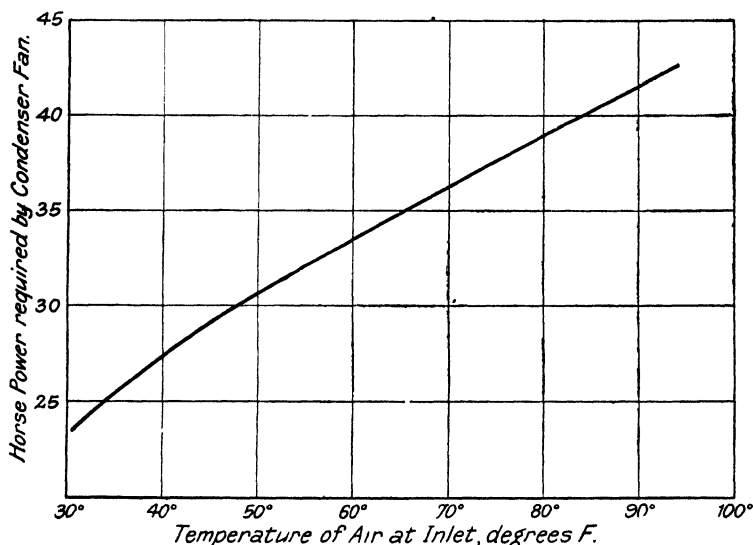


FIG 65

FAN POWER AND AIR TEMPERATURE FOR RAMSEY CONDENSER

tubes are now stationary and upright, the steam entering by flange 2 and passing down the tubes. The water enters by connections 28 and 27, and reaches chamber 26, from which it is distributed over the tubes by one of several devices. Surplus water passes back to the tank by outlet 33. In Fig. 66 a fan is shown at the front of the condenser, but experience has shown that it is better to leave the air inlet of the condenser unimpeded. In the later designs the fan is at the rear of the condenser, and assists the draught created by the motion of the locomotive to draw the air through the condenser. The fan may be driven in a turbo-electric locomotive by a motor, and in a geared turbine locomotive by an auxiliary single wheel turbine.

In its combined use of air and water for cooling, this condenser is radically different from the Ljungstrom Locomotive Condenser,

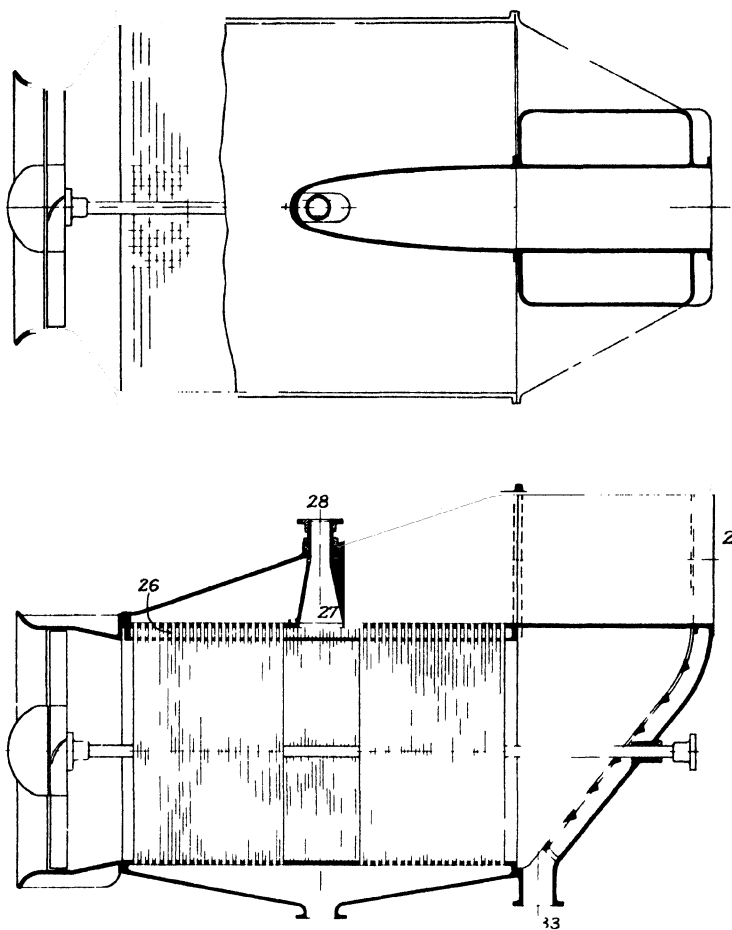


FIG. 66
LOCOMOTIVE EVAPORATIVE CONDENSER
(G. I. Jones)

which relies upon three large fans to force sufficient air over a surface condenser to maintain a vacuum *

* See *Engineering* Aug 1922

CHAPTER XIII

JET CONDENSERS

General Considerations and Types. The term " Jet Condenser " is usually applied to all forms of steam condensers where condensation is effected by direct contact of the steam with the cooling water.

The absence of cooling surface, that is, of the condenser tubes, together with the ferrules and tube plates which constitute a considerable proportion of the first cost of a surface condenser, entails a substantial saving in the first cost of a jet condenser over that of a surface condenser, and this factor generally represents the primary advantage in favour of a jet condenser. In some cases, the nature of the available circulating water is such as to make its use in a surface condenser impracticable or, at best, excessively troublesome, and under such conditions a jet condenser can again be adopted to advantage. On the other hand, their useful field of application is limited by certain disadvantages, which may be classified under three distinct headings.

LOSS OF CONDENSATE. First and foremost amongst these is the loss of the condensate as such, and this rules them out of the whole field of marine work.

Where the circulating water is sufficiently pure for use as boiler feed, this disadvantage practically disappears, providing that the condenser discharge is maintained at a temperature approaching the vacuum temperature. This condition is readily met in some types of jet condensers, whereas, as will be seen later, this is not the case in others, and under such conditions the additional cost of heating the feed must be set against the advantage in first cost in considering the relative merits of the types of plant to be considered. Although it would no doubt be possible to design a jet condenser in such a manner as to ensure that the outlet water should be as free from air as is the condensate of a surface condenser, this is not generally found to be the case, and this factor has therefore to be considered as a further disadvantage due to the loss of the condensate.

Conditions, again, may occur where the circulating water is unsuitable for boiler feed, but where an alternative limited pure water supply is available or can be purchased, and we have then to strike a balance between the saving in first cost of the jet plant

and the whole expense of obtaining feed water of equivalent value in the matter of purity and temperature to that of the condensate drawn from a surface condenser.

POWER CONSUMPTION. The second important drawback of a jet plant is that arising from the greater power consumption and capital cost of the auxiliaries. Owing to the intimate mixing of water and steam, it is possible to work with a considerably smaller difference d between water outlet temperature t_2 and vacuum temperature t_o , with the result that the water quantity may be less than that required in a surface condenser of equal capacity. No hard and fast rules can be laid down in regard to the relative heads on the circulating pumps for the two types. It follows, therefore, that as far as the circulating pumps are concerned, the power consumption may be greater or it may be less, according to the circulating water levels and the type of jet condenser under consideration. The absence of the condensate pump is naturally a point in favour of a jet plant. On the other hand, since the whole of the circulating water is exposed to the full vacuum and carries a large amount of air which is liberated under vacuum, the air pump capacity of a jet condensing plant must necessarily be greater than that of a surface plant. Naturally, the higher the vacuum to be obtained the greater the difference in air pump capacity and power consumption, and whereas this disadvantage is usually immaterial for low vacuum plants required for reciprocating engines, it becomes one of great importance in the case of high vacuum turbine condensers.

SHAPE. Lastly, there are the questions of shape and of safety. The general nature of the design of a jet condenser involves more headroom than does that of a surface condenser of equal capacity, and it is only possible to obtain the very desirable arrangement of the exhaust inlet opening at the top of the condenser by sacrificing the advantages due to the counter-current principle. In certain jet condenser types, it is necessary to provide safety devices to prevent the water rising and damaging the prime mover in case of failure of the auxiliaries.

These drawbacks are again more pronounced in the case of high vacuum turbine plants than in that of reciprocating engine installations.

As a general rule, it may therefore be assumed that the choice as between surface and jet plants can be largely decided in the case of low vacuum plants on the question of the importance or otherwise of the saving of the condensate, whereas the question of air pump capacity and convenience of the layout are frequently

factors of equal importance where the choice applies to a high vacuum plant required for a turbine installation.

Types of Jet Condensers. Jet condensers may be divided into those that are provided with a separate air pump, and those which combine the functions of a jet condenser and of an hydraulic air pump, and are generally known as ejector condensers.

We have further to differentiate between low-level and barometric jet condensers.

THE LOW-LEVEL TYPE. In a low-level plant the vacuum within the condenser draws the cooling water into the condenser, and a so-called tail pump draws the water from the condenser and delivers it to the point of discharge.

Where the suction head due to the vacuum is insufficient to overcome the sum of the static head between the water supply level and the inlet branch, and the head due to friction in the pipe line and within the condenser, then the difference has to be made up by a pump on the inlet side, usually referred to as a head pump.

A reference to any of the illustrations of low-level condensing plants will show that, in the event of the tail pump failing to act, the water will rise and may readily be drawn back into the prime mover. To obviate this danger, it is essential to fit some form of vacuum breaker that comes into action when the water level reaches a predetermined height.

THE BAROMETRIC TYPE. In the barometric type the tail pump is not required, as the condenser is placed at a sufficiently high level above the point of water discharge to enable the water to drain away from the condenser, by gravity, through a sealed tail pipe.

A barometric plant is free from the element of danger inherent to low-level plants, since the failure of either of the pumps in such a plant does not cause the water to flow back through the steam inlet.

If we consider the water level in the tail pipe, it will be seen that the height above the outlet level is determined by the vacuum in relation to the barometric pressure of the atmosphere. If, therefore, the vacuum rises in the condenser, the water level in the condenser will rise, and vice versa. If the variations in vacuum are of a sudden nature, the water column will pulsate in the tail pipe, and may shoot back into the steam space and may so reach back to the prime mover. It is therefore advisable to install a non-return valve at the foot of the tail pipe, as this will prevent the occurrence of such pulsations.

Barometric condensers have usually to be placed at a distance from the prime mover, and therefore call for more or less lengthy exhaust steam pipe lines and a certain drop in pressure across these pipes. Their use is therefore confined to reciprocating engines and to small sizes of steam turbines.

Counter Current and Parallel Flow Jet Condensers. We have next to consider the relative advantages of counter current and parallel flow jet condensers. In a counter current condenser, the temperature of the air and the amount of steam drawn over with the air are independent of the relation between water outlet temperature t_2 and steam temperature t_o , and the air passes through consecutive layers of cooling water and separates from the cooling water at the water inlet level, so that its temperature is reduced almost to the temperature of the water inlet t_1 .

In a parallel flow condenser the air separates from the cooling water at a point of maximum water temperature t_2 . For reasons already fully investigated in Chapter VI, this means that the water outlet temperature t_2 must be kept well below the steam temperature t_o , so that in addition to necessitating an increased air pump capacity, we have to reckon with a higher figure for d , and consequently, a greater water quantity in the case of a parallel flow than in a counter-current jet plant. In the case of counter current condensers, test results have shown that the maximum possible vacuum can be obtained in practice for a given water quantity, and inlet temperature corresponding to $d = 0$. From our considerations in Chapter VI, it follows that such a condition is impossible of attainment in a parallel flow plant, as this would mean that the whole body of the steam would pass into the air pump.

The important advantages of the counter current principle in jet condensers were fully recognized by Weiss in his treatise on the subject,* at a time when the turbine industry was still of secondary importance, and the principle was almost universally adopted at the time. With the increasing demand for high vacuum condensers for steam turbines, however, the problem of building condensers with exhaust openings at the top, and which could be bolted immediately on to the exhaust flange of the turbine, came to the front. In all but the smallest size turbine plants, the loss of vacuum due to exhaust pipes with rectangular bends is quite inadmissible. Unfortunately, it is not easy to combine the top exhaust steam inlet with a counter current design, and later designers have therefore concentrated their efforts on devising special means for

* *Weiss Kondensation*, Julius Springer, Berlin.

overcoming the inherent disadvantages of the parallel flow principle.

METHODS OF OVERCOMING DISADVANTAGE OF PARALLEL FLOW TYPE. A reference to the various designs illustrated below will show that the methods adopted for this purpose can be classified under two headings, namely, those consisting of an air cooling device, and those which aim at entrapping part of the air in the injection water and, frequently, at the same time partially compress the air within the condenser. Condensers embodying the last-named idea are not pure jet condensers in the narrower sense, but rather a combination of jet and ejector condensers. Such condensers are generally divided into two chambers. The upper chamber acts as the condensing vessel proper, and ends in a cone through which the water passes into the lower chamber.

The tail pump draws the water and the air pump draws the free air from the lower chamber, and since the water velocity in the cone is kept high enough to cause a partial compression of the air, the duty of the air pump is relieved by a corresponding amount. Further, the weight of air to be handled by the air pump is reduced.

In every jet condenser the water has to be finely subdivided to present the largest possible surface to the incoming strain, with the result that the water is thoroughly deaerated, or in other words, practically the whole of the air carried over by the injection water is released. This air, as well as that carried over by the steam, is entrapped in the cone, and only a fraction of the air quantity is set free again in the lower chamber during the short space of time that the water remains within the chamber. If the pressure in the upper chamber is P_u , and in the lower chamber $P_l + \Delta P$, then the temperature of evaporation is t_o in the upper chamber and $t_o + \Delta t$ in the lower chamber. It has been shown that the air temperature at the air pump inlet t_{ca} must be lower than t_o , but the arguments leading to this conclusion were based on the usual conditions prevailing in a condenser where the air pressure at the air pump inlet is equal to or lower than that prevailing at the steam inlet. Where, however, as in the design now under consideration, the pressure is higher at the air pump inlet, we have only to ensure that the air temperature at the air pump inlet is below that corresponding to the pressure there prevailing, that is to say, t_{ca} must be below $t_o + \Delta t$, but not necessarily below t_o . It follows, therefore, that the general objection to raising the water temperature t_2 to t_o in a parallel flow jet condenser does not apply in this instance, and, in point of fact, test results obtained on such condensers have

shown values for d of 2° or 3° F. The substantial advantages that accrue from the adoption of this two-stage design involve a high water velocity in the injection nozzles, which in turn increases the internal water friction head and limits the suction lift of the condenser. In most layouts, however, the conditions are such that part of the available suction head would be wasted if it were not utilized in this manner.

CHAPTER XIV

JET CONDENSERS—CALCULATION OF WATER HEAD

Water Head on Head Pump. Taking first the case of low-level condensers, we have a suction head on the inlet water side equal to the vacuum prevailing in the condenser. The whole of this head is not usually available for raising the water in the inlet branch, as part of the head is mostly employed internally in forcing the water through nozzles within the condenser. The actual amount so employed depends on the individual design, but as a general rule, a suction head of the order of 20 ft. is available, and in most layouts this head is ample to overcome the static and friction head on the inlet side. If the water supply is at a higher level than the exhaust flange of the prime mover, that is to say, if the water is fed under pressure, then it is not safe to feed directly into the condenser, as, in the event of a failure of the tail pump, there would be no effective means of preventing the water from flooding the condenser and rising into the exhaust end of the turbine or the low-pressure cylinder of the engine. An automatically operated water sluice valve, designed to close in the event of the water rising in the condenser, due to any cause, would not be sufficiently reliable in its action, as a slight leakage through such a valve would lead to flooding in time. The only safe way of dealing with such conditions is to arrange for the water to feed into a tank provided with an overflow and open to the atmospheric pressure, and whose water level is below that of the condenser inlet branch. The condenser can then draw its supply of water from the tank so long as a vacuum prevails. As soon as the vacuum is broken in the condenser the water will naturally cease to rise to the inlet branch level.

In many cases the available suction head exceeds the amount required for overcoming the static and friction head on the inlet side of the condenser; in other words, the available head across the nozzles will exceed that required for obtaining the required water velocity within the condenser. The surplus head will then be wasted. It would, of course, be possible to utilize this surplus head by inserting a water turbine in the inlet pipe line, and arranging for this to provide part of the power required by the tail pump and air pump, but the capital cost of such an arrangement would only be justified under exceptionally favourable conditions.

The head on the inlet or head pump of a barometric condenser is determined by the vertical distance from the supply water level to the centre line of the condenser inlet branch, less the available suction head, due to the vacuum in the condenser.

Water Head on Tail Pump. The tail pump head is made up of the head required to draw the circulating water and the condensate from the condenser, that is the head corresponding to the difference in pressure prevailing within and without the condenser, plus the static and friction head on the delivery side. The static head is determined by the difference in levels of the water within the condenser and the point of discharge. The discharge level may be below the level in the condenser, and providing the discharge is sealed, the resultant negative head can be treated as a negative quantity in calculating the head. If the negative head is equal to the vacuum head, plus the friction head in the discharge pipe, then no tail pump is needed, and we have the conditions of a barometric condenser.

To put these considerations into the shape of formulae—

Let P_t = Total head on head pump.

P_i = Total head on tail pump.

P_v = Head corresponding to difference in pressure within and without the condenser.

P_s = Difference in water level between point of circulating water supply and centre line of water inlet branch on the condenser.

P_{if} = Friction loss in the inlet pipe line.

P_o = Difference in water level between the level of the water in the condenser and the point of water discharge.

P_{of} = Friction loss in the outlet pipe system.

P_n = Head across nozzles and internal water friction loss.

All the above to be assumed to be in the same units.

Then taking first the case of the head pump. In a low-level plant this pump is not required if

$$P_v = P_n + P_s + P_{if} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

If P_v exceeds the sum of the items on the right side of the equation (1), then a head pump has to be used, and

$$P_t = P_n + P_m + P_{if} - P_v \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Equation (2) gives the head on the head pump of a barometric condenser. P_v should be taken as the figure corresponding to the

lowest vacuum at which it is intended to maintain the condenser in operation.

For the tail pump of a low-level condenser

$$P_t = P_v + P_o + P_{of} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

In this instance P_v should correspond to the highest vacuum that is likely to occur under any conditions of load or of water temperature. As already stated, P_o , if a negative quantity, can only be inserted in (3) if the discharge is sealed. If the discharge is open and P_o is negative and equal to or greater than P_{of} , then we can take

$$P_t = P_v \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The height of the tail pipe of a barometric condenser is given by

$$P_o = P_v + P_{of} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where P_v should again correspond to the maximum vacuum.

CHAPTER XV

JET CONDENSERS—TYPICAL DESIGNS

Metropolitan-Vickers Jet Condenser. Fig. 67 illustrates a low-level jet plant built by the Metropolitan-Vickers Electrical Co., Ltd., of Trafford Park, designed for maintaining a high vacuum for turbine purposes and with the exhaust opening *A* at the top.

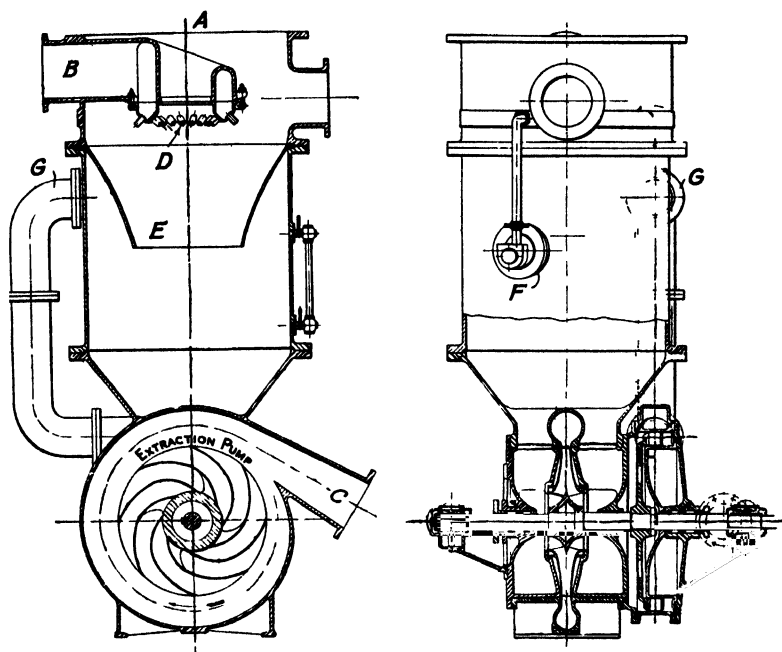


FIG 67

METROPOLITAN-VICKERS LOW LEVEL JET PLANT

The injection water is admitted at the opening *B*, and enters a water distributing chamber *C*, from which it passes into the condenser proper through a series of nozzles *D*. The nozzles are of comparatively large area so as to prevent them being readily choked by foreign matter, and they are so designed that the injection water presents a very large surface in contact with the steam. The water and steam then enter the cone *E*, and steam and water are intimately mixed so that complete condensation is effected. The

makers allow a considerable drop in water pressure across the nozzles, and the resultant water velocity is utilized in the cone *E* for partially compressing the air so that the duty devolving upon the air pump is reduced. Since the water falls from the cone in an almost unbroken stream into the tail pump, a substantial proportion of the air is, no doubt, held entrapped by the water, and in this respect, too, the air pump is partially relieved, and the inherent difficulties of the parallel flow design are largely neutralized. The air which separates from the water passes through pipe *G* into the air pump.

As will be seen from the illustration, the tail pump and the air pump are mounted on a common shaft and the casings of the two pumps are bolted together. In small plants the two pumps are built into a single casing. It will be seen that the tail pump casing forms the lower part of the condenser. Brackets are cast on the side of the condenser at a higher point, and these in turn are supported on horizontal girders let into the turbine foundations, without throwing the weight of the condenser on either the turbine or on the extraction pump.

In other designs the condenser body is carried down to the floor level, and the tail pump opening is arranged in the side of the lower portion of the condenser, and the condenser and the pumps are bolted to the floor. The air pump shown is of the Leblanc rotary valveless type. This type of pump provides a safeguard against any possibility of the water rising into the turbine exhaust in the event of the air pump failing to act, as there is a clear way through the pump, and, in case of its failure, the air rushes back through the air pump and breaks the vacuum in the condenser. In addition, a float controlled vacuum breaker *F* is fitted in case the extraction pump should fail, and the air pumps continue in service.

The M.A.N. Jet Condensers. The jet condensers built by the Maschinenfabrik Augsburg Nürnberg, are of the parallel flow type with the exhaust steam opening at the top, and are provided with a special air cooling arrangement. The type of nozzle employed is a peculiar feature of the design, and is intended to be proof against choking in the event of foreign matter being carried over with the circulating water.

As will be seen from Fig. 68, the water is admitted to an annular belt *a* and passes from this into concentric ring-shaped nozzles *b* of inverted egg section that are open at the bottom. The water pours through these openings in the form of two or more concentric sheets of water which impinge on concentric guide plates *c*. The bottom edges of these guide plates are turned up to form settling

channels for the water. The water pours over the edges of these channels in a further series of concentric sheets and collects at the bottom of the condenser body, which forms a hot well and passes from there into the tail pump. An auxiliary cold water supply enters the annular channel *d* and overflows the edge so as to form a cylindrical sheet of cold water through which the air has to pass before it can enter the air pump suction. The steam and air paths are indicated by arrows in the illustration. It will be seen that the lower sheets of water form an ejector which entraps the steam and air, and ensures complete condensation. The company use their hydraulic air ejectors—Dr. P. H. Müller's patent—in conjunction with their jet condensers.

The Alberger Jet Condensers. The Alberger Pump and Condenser Co., of New York, build two distinct types of low-level jet condensers, namely, a parallel flow design with the steam opening at the top, and a counter current design with side ways exhaust opening.

Fig. 69 shows a condenser of the former type mounted immediately over the tail pump whose casing is strengthened to support the weight of the condenser. The condenser body is of cast iron, and where desirable it can be carried down to floor level, the lower part forming a hot well, and the tail pump opening is then bolted to a vertical flange. The inlet water enters an annular channel and passes into the steam space through suitably designed helical spray nozzles, which break the water into a finely divided state, and brings it into intimate contact with the incoming steam. The mixture of cooling water and condensate passes through a cone in which the air is entrapped in the water as far as possible, and then falls in a solid stream through the lower part of the condenser body

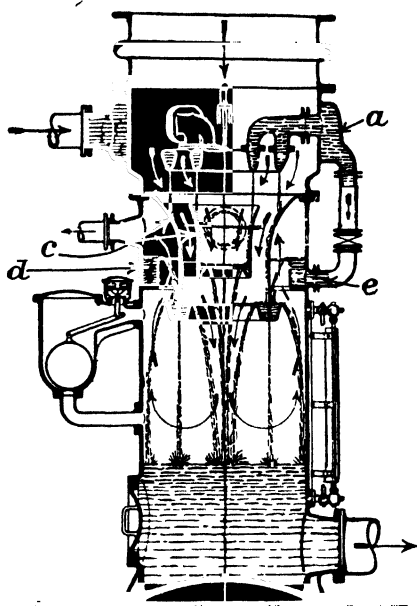


FIG. 68
M A N. JET CONDENSER.

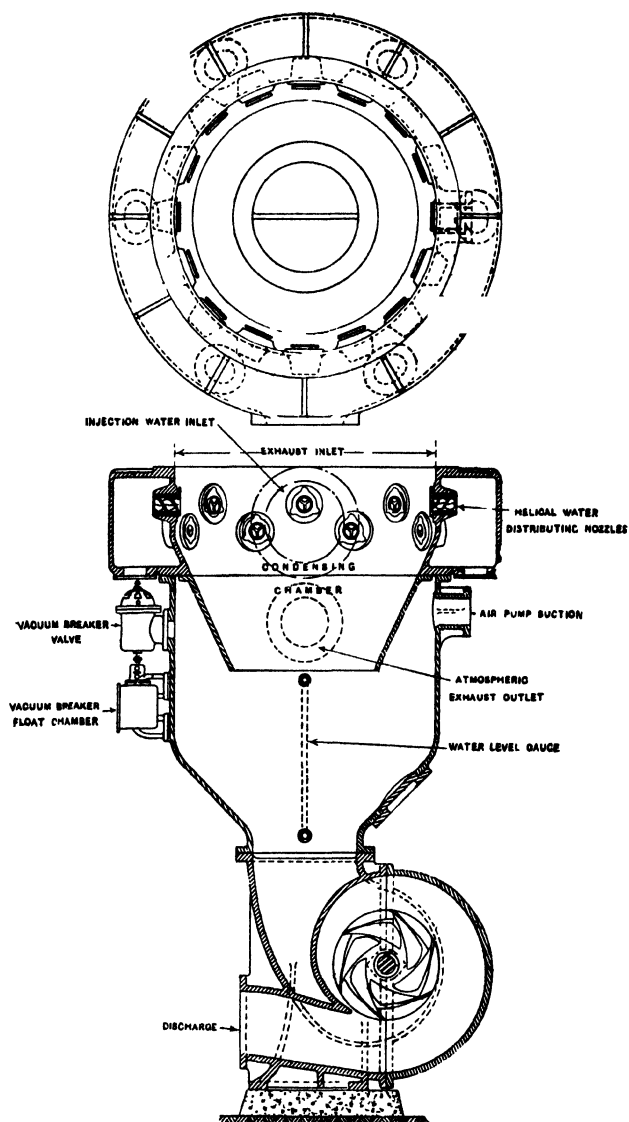


FIG. 69 (see p. 149).
ALBERGER PARALLEL FLOW JET CONDENSER.

into the pump opening. The air suction branch can be seen in the illustration and is connected to the highest point of the lower part of the condenser.

To prevent the water rising above a predetermined height, the makers provide their improved vacuum breaker. This consists of two parts—one, a float chamber in which a float operates a small pilot valve. This in turn admits air through a small pipe connection to one side of a piston which operates the main breaker valve in a valve chamber. This arrangement permits the use of a much smaller float than is necessary if the float controls a main breaker valve direct, but its greatest advantage is claimed to be due to the fact that, instead of destroying the vacuum completely and causing the main power unit to exhaust to atmosphere, the vacuum is only reduced by a few inches or sufficiently so to decrease the quantity of incoming injection water, and to assist the tail pump in discharging the accumulated excess water. The moment this occurs the breaker valve closes and the full vacuum is restored automatically.

The tail pump shown is of a special design which the makers claim to have been originated by them. The casing is vertically divided, and is so arranged that the back part of the casing can be removed without disturbing the condensing cone which is mounted above the pump, the pipe connections, or the alignment. The whole rotating element can be removed and replaced without difficulty. The discharge opening is below the centre line of the pump and points in a horizontal direction. The suction opening points vertically upward, and is so designed as to permit mounting the condenser body above the pump.

Sealing rings are provided around the impeller hubs in the pump casing. They are designed effectively to prevent leakage and are readily renewable.

A deep stuffing box is provided at either side where the shaft passes through the casing, and each box is water-sealed by means of a bronze lantern gland which must be connected to a source of clear water supply under moderate pressure.

The impeller is of the enclosed type with double entrance openings, and designed to operate in hydraulic balance and to eliminate end thrust.

Two pedestal bearings are provided independently of the pump casing, one on either side of the pump, separated by the stuffing boxes from the water being pumped. They are of the ring-oiling type, with oil reservoirs and oil sight gauges. They are provided with removable caps and split bushings, so that they can be dismantled without disturbing any other part of the pump. A

multi-collar marine type thrust bearing with an adjusting device is incorporated in the outboard bearing to maintain the rotating element in correct lateral position.

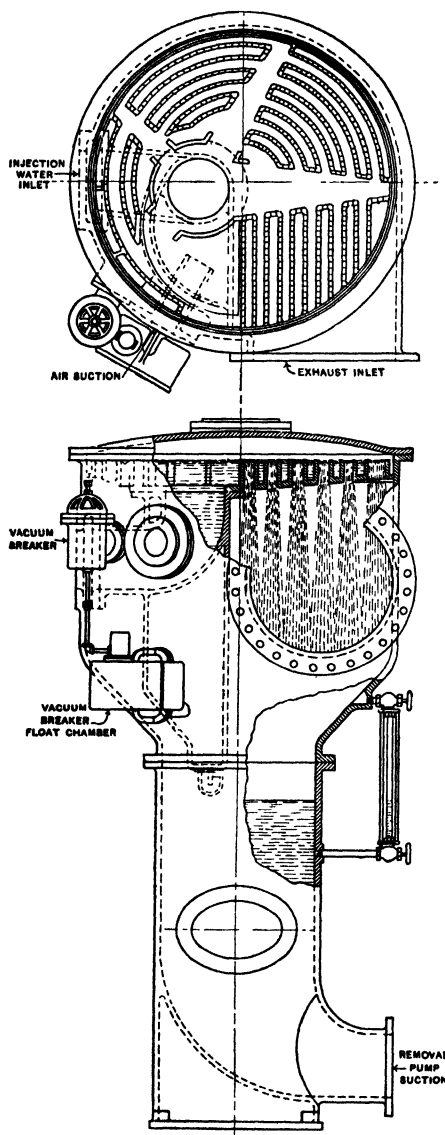


FIG. 70.

ALBERGER SPIROFLO JET CONDENSER.

THE ALBERGER SPIROFLO CONDENSER. Fig. 70 shows a sectional view of a side exhaust condenser, known as the Spiroflo type, made by the Alberger Co. In this design the counter current principle is followed as far as it is practicable to do so without arranging the air outlet in the top, which would involve a considerable increase in head room. The present design, however, is particularly intended to reduce the required head room to a minimum. The exhaust opening is placed tangentially to the main body. A vertical partition plate is so arranged in the interior as to provide a spirally-shaped chamber of gradually decreasing section, the large end terminating in the steam inlet branch and the air being withdrawn at the small end.

The cooling water feeds into a series of shallow cast iron trays mounted above the condensation chamber. These trays are designed to give a very complete distribution of water spray

throughout the condensing chamber, and at the same time the openings between the trays are large enough to prevent their being stopped up by foreign matter carried over by the cooling water. In the present illustration the condenser is mounted on a hot well, and the tail pump is to be bolted to the vertical outlet flange.

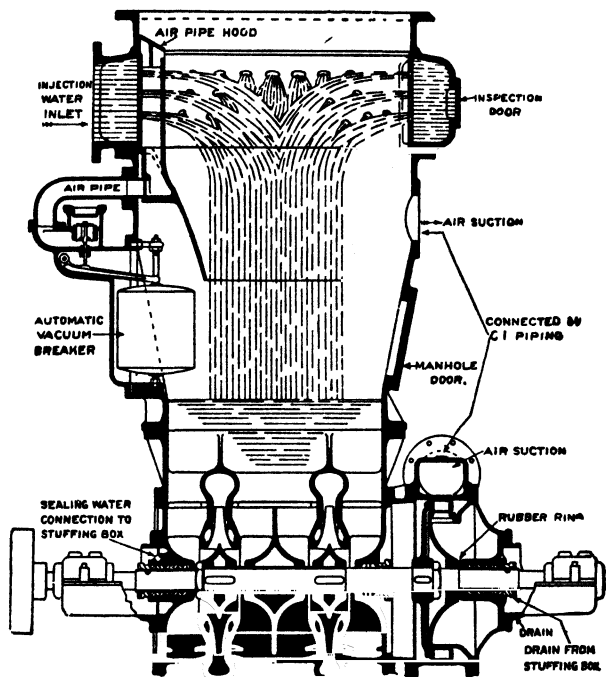


FIG. 71 (see p 154)

MIRRIE'S WATSON LOW-LEVEL JET CONDENSER

THE ALBERGER BAROMETRIC CONDENSERS. The Barometric Condensers built by the Alberger Co. are provided with a condensing chamber which is similar in design to that of the Spiroflo low-level condenser. The tail pipe is of cast iron, made in several sections with flanged joints. The bottom section is provided with a bracket for support on the foundations, and the whole tail pipe construction is strong enough for the support of the whole structure. The company make two types of air pump for use in connection with their jet condensers, one of the dry horizontal reciprocating type with positively actuated slide suction valve and poppet type discharge valves, the other of the two-stage steam ejector type with jet or surface intercooler and known as an Air Occluder.

The Mirrlees-Watson Jet Condenser. Fig. 71 illustrates a low-level multiple jet condensing plant built by Messrs. The Mirrlees-

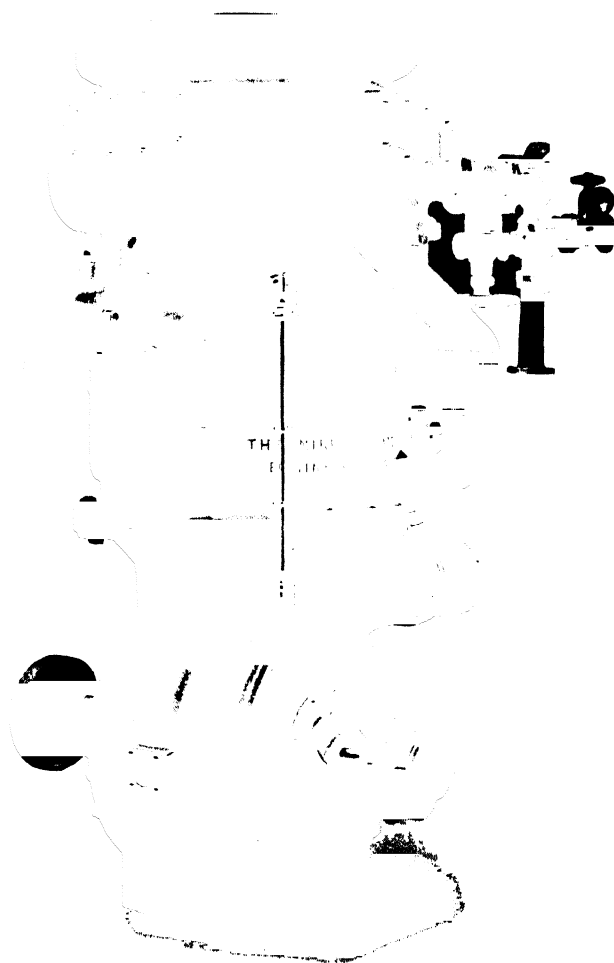


FIG 72
MIRRLEES-WATSON LOW-LEVEL JET CONDENSER WITH STEAM
JET EJECTOR.

Watson Co., Ltd., of Glasgow, and provided with a Mirrlees-Leblanc rotary air pump.

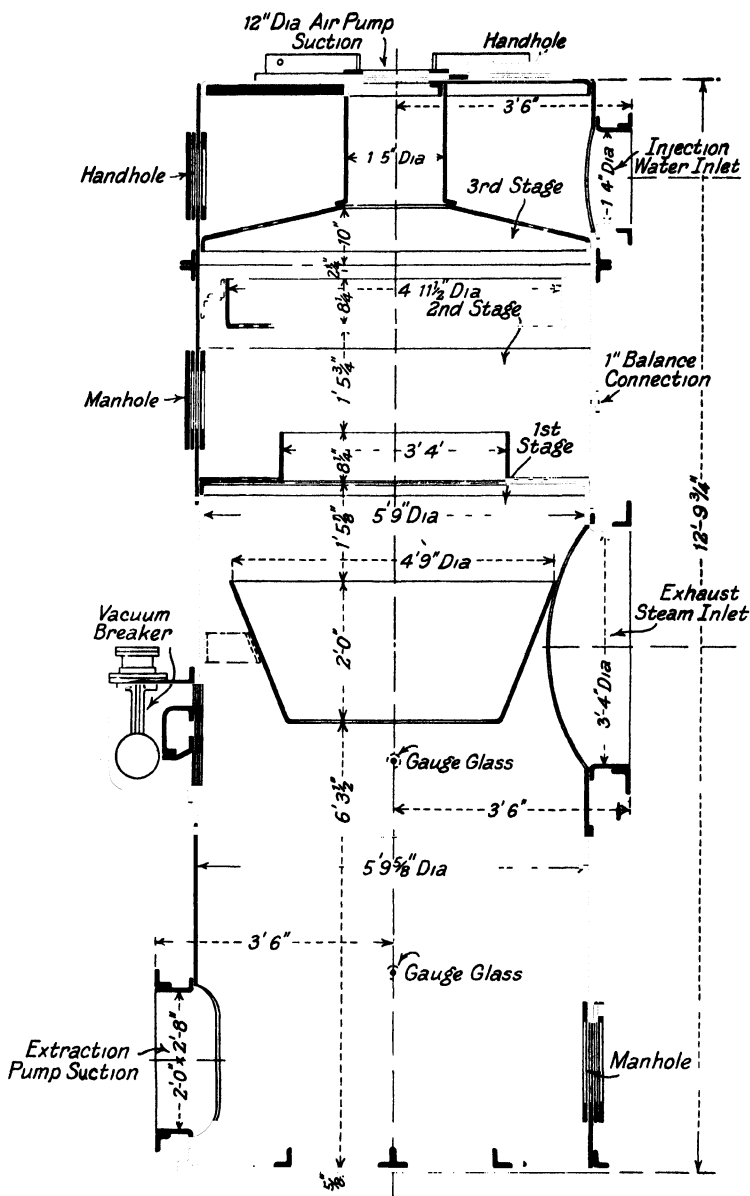
The condenser is built on the parallel flow principle. The steam

enters at the top of the condenser so that the exhaust flange of a turbine can be bolted directly on to the condenser. The water passes round an annular space outside the main body of the condenser and enters the steam space through nozzles arranged in three tiers. The kinetic energy of the incoming steam and water is utilized for partially compressing the air in the compression cone. The cast iron body of the condenser is fitted with lugs to support it on horizontal girders, and the tail pump casing is underslung and bolted to the bottom flange of the condenser body. The arrangement is clearly shown in Fig. 72, illustrating a similar condenser with a steam air ejector in place of the rotary air pump. The water extraction pump is of Messrs. Mirreles-Watson's design and make, and their extensive experience in the production of this class of pump has enabled them to produce a design in which the risk of corrosion is reduced to a minimum. As will be seen, the pump has two impellers which work in parallel and is arranged for direct drive by means of an electric motor, the air pump impeller being mounted on the same shaft and the two pumps forming practically a single unit.

The float-controlled vacuum breaker is so arranged that the incoming air that breaks the vacuum and interrupts the injection water supply enters the condenser at a point above the water inlet. This arrangement ensures the reliability and the quickness of action of the device. The efficiency of the design as a whole is proved by the fact that the makers are able to work with a cooling water quantity based on $d = 4^{\circ}$ F. Plants of this type have been made for duties up to 200,000 lbs. of steam per hour.

Worthington-Simpson Jet Condenser. The sectional arrangement shown in Fig. 73 illustrates a low-level counter current condenser built by Messrs. Worthington-Simpson, Ltd., of London and Newark, with the steam inlet at the side. This condenser is designed to deal with 46,500 lbs. of steam per hour, and to maintain a vacuum of $27\frac{1}{2}$ ins. when supplied with 3,620 gals. of cooling water per minute at a temperature of 80° F.

The purpose of the inverted cone-shaped guide opposite the steam opening is to ensure an even distribution of the steam over the whole circumference of the condenser. The water enters a water chamber at the top of the condenser, arranged concentrically round the air outlet, and passes downwards through a series of three perforated trays that divide the water into a fine spray. The perforations in each tray consist of 504 one-inch holes, arranged in 6 concentric circles. The three trays divide the condenser into three stages. In the first or bottom stage there is a central core, formed by the water



Section through Condenser

FIG 73 (see p 155)

SECTIONAL ARRANGEMENT OF LOW-LEVEL COUNTER CURRENT
JET CONDENSER
(Worthington Simpson.)

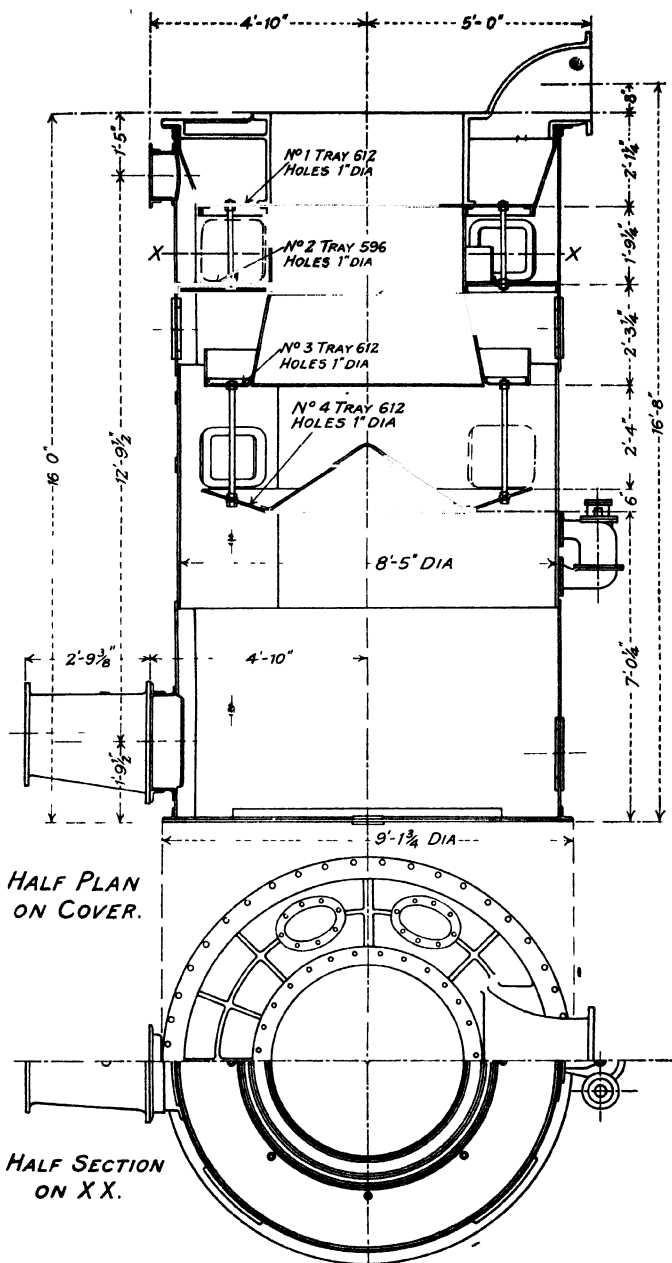


FIG. 74 (see p. 158).
 WORTHINGTON-SIMPSON COUNTER CURRENT LOW-LEVEL JET
 CONDENSER.

jet passing through the inside of the inverted cone, and the steam has to pass through the water spray falling from the perforations in the bottom tray and through the central opening in the tray to the second stage. It then flows outwards through the spray formed by the intermediate tray and through the annular space formed round this tray into the top stage, where the remnant and the air pass inwards through the third spray towards the air outlet.

WORTHINGTON-SIMPSON CONCENTRIC DESIGN. Fig. 74 shows a condenser of the same firm's manufacture in which the exhaust opening is at the top, and where the counter current principle is fully maintained by arranging the condenser space proper concentrically round a cylindrical steam passage that guides the steam to the bottom stage of the condenser, with the result that the steam first comes in contact with the hottest water and then passes upwards through a succession of water sprays towards the air outlet. In this condenser the water passes over four perforated trays. Suitable baffle plates are provided and are indicated in the drawing. They ensure that the steam is evenly distributed circumferentially and guard against short-circuiting of portions of the water sprays. This condenser is designed for a normal steam quantity of 53,000 lbs. per hour, to maintain a vacuum of 27 ins. when supplied with 5,000 gals. of cooling water at a temperature of 90° F.

The condenser shell is built of $\frac{7}{16}$ in. steel plates. The water inlet branch leads to an annular water chamber formed in the upper section of the condenser, and whose bottom forms the first water tray. The tail pump draws the circulating water through a 20 in. branch on the side of the shell close to the bottom of the condenser.

The Hick-Breguet Low Level Jet Condensers, made by Messrs. Hick-Hargreaves, of Bolton, are of the multiple jet parallel flow type, as illustrated in Fig. 75. The injection water is led into a closed annulus in the upper part of the condenser in which spray nozzles marked *R* of a very simple form are located, the number of these nozzles being accurately proportioned to the duty of the plant.

The steam inlet, which is at the top of the condenser, is made of ample proportions for maximum duty.

The upper portion of the condenser is designed to reduce as far as possible the resistance to the passage of the steam on to the finely divided sprays of the water jets, condensation in this way being rapid and complete.

The water of condensation and the injection water fall to the bottom of the condenser, where they are withdrawn by means of an extraction pump marked *F* of the centrifugal type with

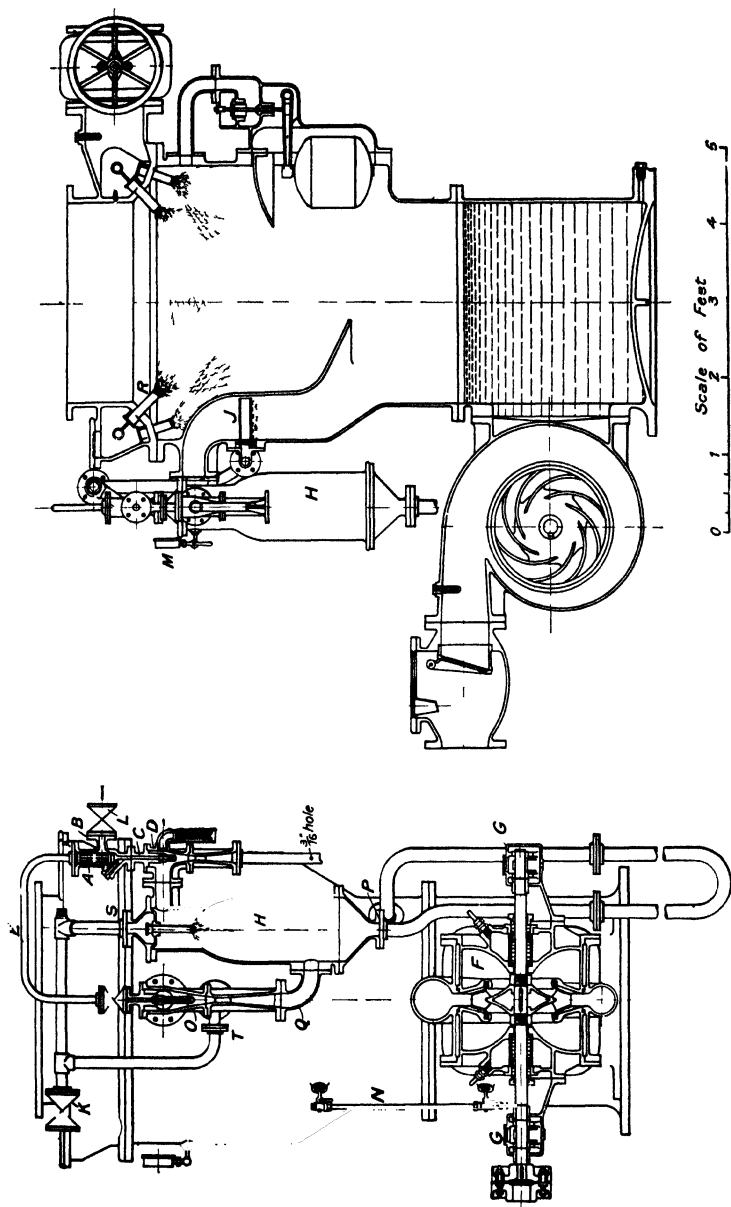


FIG 75.
HICK-BREGUET LOW-LEVEL JET CONDENSER.

ring lubricated bearings marked *G* and which is operated by means of an electric motor or mechanically as required. Should the means of operating the pump fail, the injection water would rise in the condenser and bring into action the vacuum breaker *X*, which instantly dissipates the vacuum in the condenser and prevents any further flow of injection water.

The air is withdrawn separately by means of an "Ejectair" type air ejector marked *OHD* from immediately above the surface of the water in the bottom of the condenser. The air is increased in density by being cooled by a water jet *J* connected to the main injection pipe, the air being withdrawn at a temperature corresponding approximately to the inlet temperature of the injection water. The water from the auxiliary condenser along with the condensate from the first jet is returned by means of a *U* balance pipe marked *Y* regulated by a diaphragm marked *P*. Diaphragms are also placed at the points marked *T* and *S*; the object of the former is to prevent an undue amount of water being supplied to the sprays to the detriment of the water supplied to the auxiliary condenser *H*. As the vacuum in the auxiliary condenser *H* is a few inches less than that in the main condenser, there is a tendency for the water to flow through the cooling sprays *R* in preference to coming into the auxiliary condenser. With this type of plant it is not usual to conserve the heat in the first jet. The loss entailed is extremely small, as this jet only uses about one-third of 1 per cent of the total steam used by the prime mover.

The steam from the second jet marked *D* may be used by discharging it into a feed heater or an open feed tank as may be preferred. It is claimed that the simplicity of this type of condensing plant represents a feature of importance; all that is necessary to put it into operation is to open the steam valve marked *L* to the nozzles and to close the switches controlling the electric motor driving the extraction pump. As the nozzles are small in bore, the steam is passed through the strainer marked *A* before passing the nozzles.

It will be noted that no lift pump is required in connection with this plant, the extraction pump being capable of lifting the water to any reasonable height such as is usually called for when cooling towers are installed.

To ensure stability of the whole apparatus, a stabilizing valve marked *Z* is fitted on to the second jet. The object of this is to keep the vacuum in the auxiliary condenser from rising too high and from breaking down the second jet.

In the case of a jet plant having the extraction pump driven

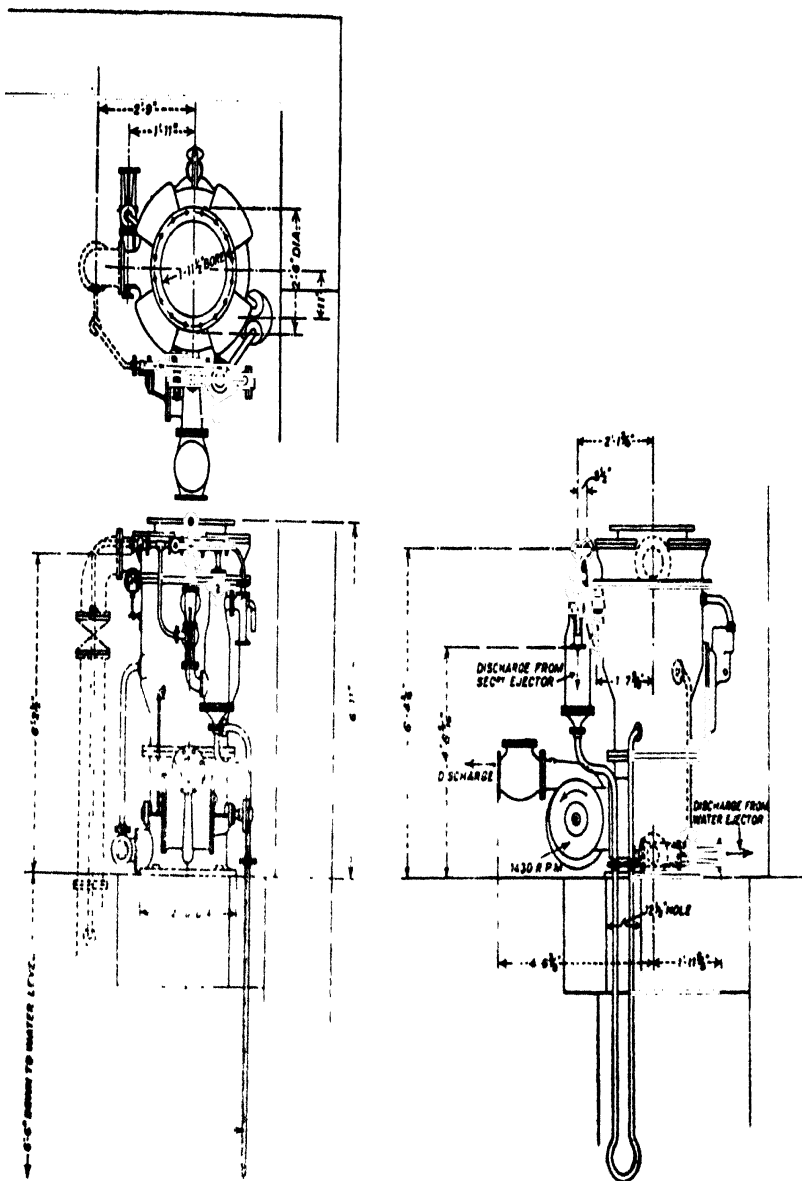


FIG. 76 (see p. 161).

GENERAL ARRANGEMENT OF HICK-BRAZOUT LOW-LEVEL JET CONDENSING PLANT.

from the engine or turbine shaft, a water extracting ejector operated by steam is usually required. The reason for this is that the extraction pump does not withdraw the water from the condenser until a certain speed is reached, the result being that water would accumulate in the condenser. To prevent this taking place, an ejector is fitted which extracts the water until the extraction pump comes into operation.

The water ejector has a capacity considerably less than that of the extraction pump, as it is intended to withdraw only sufficient water from the condenser to enable the main unit to overcome its friction load. Immediately the main unit is up to speed, the main injection valve is opened out to the desired extent.

Arrangement drawing Fig. 76 illustrates the water extractor. The plant as shown on this drawing is working under the following conditions—

Condensing 11,000 lbs. of exhaust steam and maintaining a 27 ins. vacuum (bar. 30 ins.), when supplied with injection water at 75° F.

Amount of injection water is 35,200 gallons per hr.

Steam pressure of the ejectair is 100 lbs. per sq. in., and the extraction pump operated at 1,440 r.p.m.

CHAPTER XVI

JET CONDENSERS—EJECTOR TYPE

IF we analyse the nature of the work to be performed by a jet condensing plant, it will be seen that this is of a two-fold order, namely, the delivery of the cooling water into the condenser, and its division into a number of jets or sprays designed to present a maximum water surface per unit weight of water to the steam to be condensed, and secondly, the compression of the air and of the water from vacuum pressure to that of the surrounding atmosphere.

In a jet condenser of the ejector type, the work of compression is performed by water jets formed by the injection water within the condenser body, and the only auxiliary pump required is a simple water pump which supplies the required water quantity at a sufficiently high pressure to provide the water velocity for the water jets in the condenser. As it is not practicable to subdivide the water jets in an ejector condenser to the same extent as in other forms of jet condensers, the ratio of water surface exposed per unit weight of water is naturally less favourable, with the result that designers are obliged to work with a higher value for d ; in other words, a larger water quantity is needed than in other forms of jet condensers. For this reason the theoretical amount of work to be performed by the pump will exceed the corresponding amount of work in a condenser provided with a separate air pump, but in practice the actual power consumption is also dependent on the relative efficiencies of the types of pumps employed.

The working conditions for the head pump supplying water to an ejector condenser are such as to lend themselves to a simple and efficient design of centrifugal pump, since the suction lift can generally be kept low and the duty does not fluctuate with the load, whereas the conditions in a low-level jet plant for the design of the tail pump are exceedingly difficult. It is therefore found in practice that the power consumption of an ejector plant of the multi-jet type is, at any rate, not much higher than that of other forms of jet plant where the available difference between water inlet temperature and steam temperature, that is D , is fairly high, so that the advantage or otherwise of using this inexpensive and simple form of condenser is largely determined by the degree of the vacuum to be attained for a given water temperature,

provided, of course, that the water conditions permit the use of any form of jet condenser at all.

The Koerting Ejector Condenser. The most widely adopted type of ejector condenser is that due to Messrs. Koerting Bros., of Hanover. Fig. 77 shows a Koerting Multi-jet condenser as manufactured by Messrs. Schutte & Koerting Co., of Philadelphia, U.S.A. The condenser consists of a closed cylindrical chamber, in whose upper end there is a water nozzle case to which is fitted a Venturi shaped combining tube, consisting of several sets of tapering rings joined together by equally spaced ribs. The combining tube is connected with a tail pipe discharging into the hot well. The water nozzle case is provided with a nozzle plate containing the water nozzles. The condenser body has two steam inlets, either one of which can be used. The other is closed by a flanged cover.

The injection water is forced under pressure into the water nozzle case of the condenser and passed through the water nozzles, which are designed to discharge a specified amount of water according to the conditions of flow determined by the water pressure and the existing vacuum. As the water jets emerge from the nozzles they are directed into the throat of the combining tube, where they unite to form a single jet. The exhaust steam enters the condenser chamber through the exhaust steam inlet, flows through the annular passages between the tapering rings, comes into direct contact with the

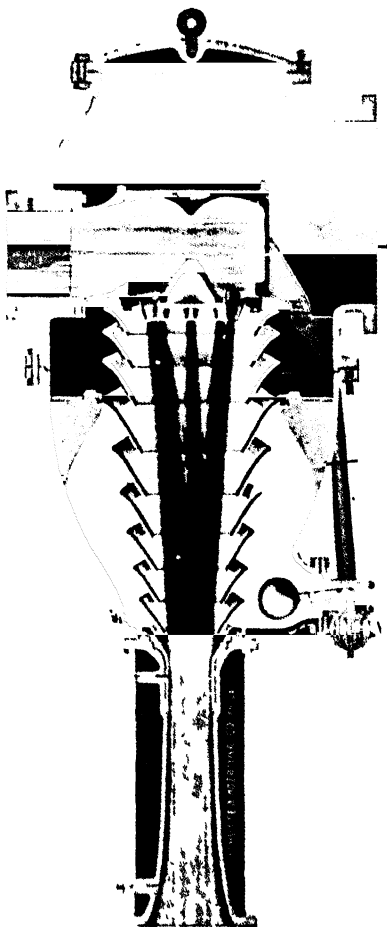


FIG. 77.

KOERTING EJECTOR CONDENSER.
(By Schutte & Koerting Co., of Philadelphia.)

converging water jets, and is condensed. Due to the combined effect of the external water pressure, the vacuum existing in the condenser, and gravity, the multiple water jets attain a high velocity that is sufficient to entrain the condensed steam, air and uncondensed gases, and to discharge them into the hot well against the pressure of the atmosphere. The water jets create a vacuum by condensing the steam, and maintain the vacuum by entraining and removing the air and uncondensed gases.

The tapering rings forming the combining tube are designed so that the sectional area of the steam passages increases from the bottom to the top, thus, since the steam in the upper part of the combining tube is in contact with the coldest water, and therefore a greater weight of steam can be condensed per unit of contact area than in the lower part of the tube where the water is hottest, the sectional area of the steam passages is practically proportional to the volume of steam flow. For this reason, the steam velocity is uniform at all points in the combining tube. Furthermore, the rings are so designed and arranged as to direct the steam flow parallel to the water flow.

The discharge end of the tail pipe is submerged about 6 ins. in the hot well and is effectively water sealed, thus preventing air entering the condenser body.

Koerting multi-jet condensers require a water supply under 9 lbs. per sq. in. pressure at the condenser injection water inlet, or water at 21 ft. head. This pressure is sufficient to overcome friction and impact losses in the combining tube and tail pipe, and to

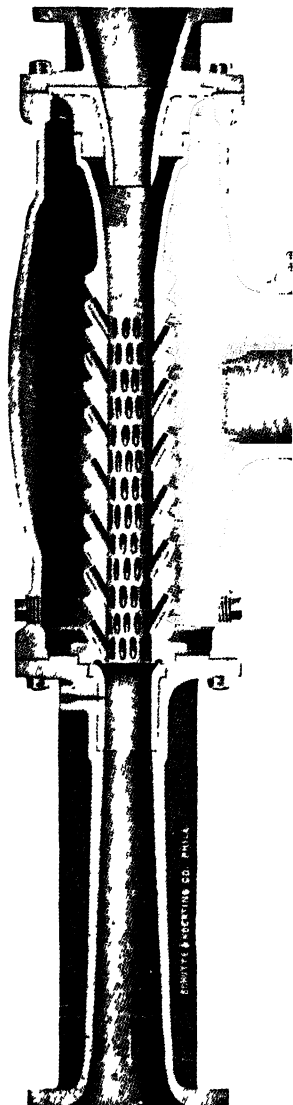


FIG. 78
KOERTING EDUCIOR
CONDENSER
(By Schutte & Koerting Co.
of Philadelphia.)

discharge the water against the pressure of atmosphere. Standard high efficiency centrifugal pumps are used for injecting the water under pressure into the condenser.

THE KOERTING "EDUCTOR" CONDENSER. The Koerting "Eductor" condenser, illustrated in Fig. 78, is designed for condensing smaller quantities of steam than handled by the multi-jet condenser. The principal difference between the two condensers is that the eductor condenser utilizes a single water jet, whereas the multi-jet condenser, as its name implies, utilizes a number of jets which unite to form a single jet in the combining tube.

Should the injection water pump for any reason fail to supply sufficient water to the condenser, the existing vacuum will cause water to pass from the hot well through the tail pipe into the condenser body. As this water rises, however, it raises the copper float of the vacuum breaker, thereby opening the breaker, admitting air into the condenser, and completely destroying the vacuum.

Another safeguard is provided by the arrangement of the tail pipe. The hot well of the condenser is divided by a weir into two parts. On the upstream side of the weir, the free discharge pipe of the condenser is submerged to a depth of about 6 ins. With this arrangement, the discharge end of the tail pipe quickly uncovers should water be drawn up into the condenser, thus breaking the vacuum and preventing water from flooding the prime movers. The discharge end of the tail pipe is cut off at an angle of 45°—a construction which permits air to enter together with the water, and which avoids any hammering effect.

In all installations a basket strainer is installed on the suction side of the water injection pump so that dirt, leaves, etc., in the injection water are prevented from entering the condenser and clogging the water nozzles.

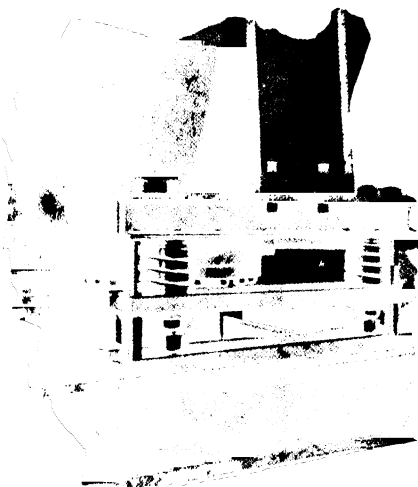


FIG. 79 (see p. 166)
SPRING SUPPORT FOR KOERTING CONDENSER.

The resultant pressure per square inch on the nozzles is equal to the combined pressure of 9 lbs. at the water inlet head and the pressure of the existing vacuum. Since the nozzle diameters vary from $\frac{1}{2}$ in. at the discharge end to 1 in. at the inlet end for the smallest nozzles, and from $2\frac{1}{2}$ ins. at the discharge end to over 4 ins. at the inlet end for the largest sizes, the total pressure is at all times sufficient to keep the nozzles thoroughly scoured.

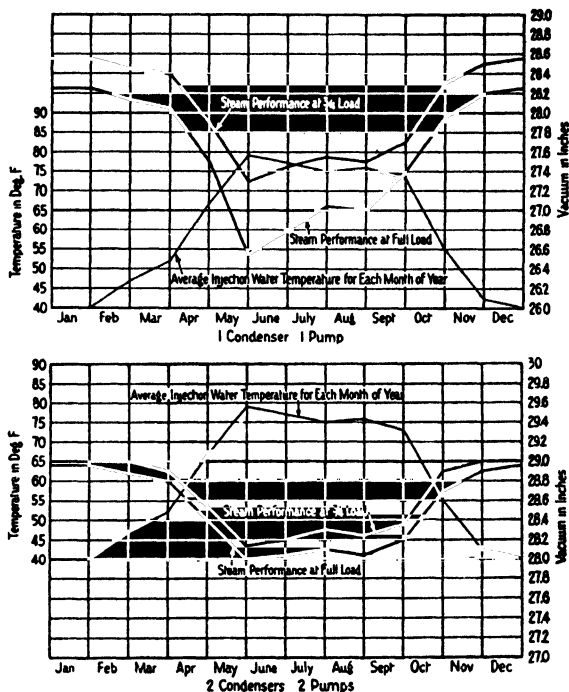


FIG 80 (see p 167)

PERFORMANCE OF TWIN EJECTOR CONDENSER
(By Schutte & Koerting Co., of Philadelphia.)

The Koerting multi-jet condenser requires about 25 to 75 per cent more water than do other types of jet condensers.

The condenser body may be bolted directly to the turbine exhaust flange and supported on springs. This arrangement decreases the head room required. An arrangement of this kind is shown in Fig. 79.

TWIN KOERTING MULTI-JET CONDENSERS. Above 5,000 kw. capacity, twin Koerting multi-jet condensers are generally installed. The advantage of the twin arrangement is great flexibility. One

pump and one-half the condensing unit are sufficient for operating under conditions of light load or cold water. With the injection pump installed in duplicate, any accident to one of the pumps will not shut down the condenser completely. The unit will continue to operate with one pump under a somewhat lower vacuum.

The hot well is generally located underneath the condenser and a straight pipe 24 ins. long is $\xrightarrow{A}$ attached to the tail pipe. This tail extension is about 6 ins. from the hot well. Except for large capacity condensers, the hot well need not be larger than 2 ft. square by 4 ft. deep.

PERFORMANCE OF KOERTING PLANT The curves shown in Fig. 80 represent the performance of a twin Koerting condensing plant at full load corresponding to a steam duty of 112,000 lbs per hour, and at three-quarter load corresponding to 84,000 lbs. per hour with variations of mean water temperature over a period of twelve months. The first set of curves gives results obtained with both condensers and both pumps in use, and the second set represents similar conditions

with a single condenser and pump in action. This condensing plant is installed at the Power Station of the Baltimore Consolidated Gas, Electric Light and Power Co., U.S.A.

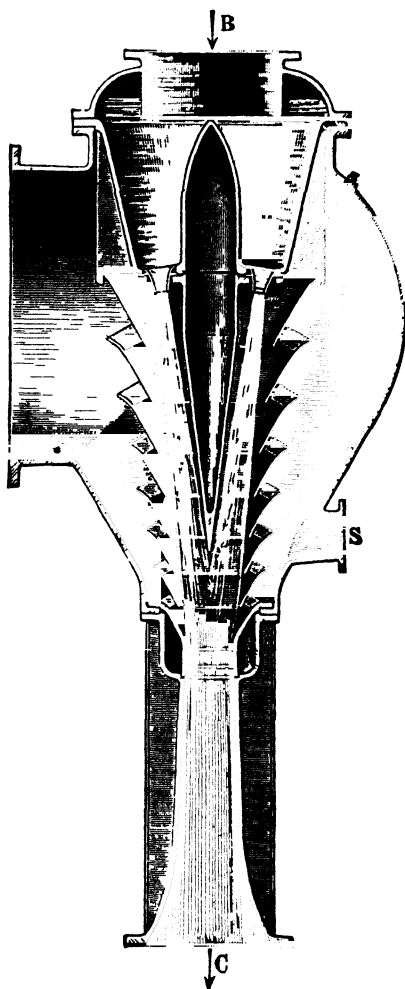


FIG. 81 (see p 168).
KOERTING EJECTOR CONDENSER
(By Koerting Bros., of London.)

The Koerting patent multi-jet ejector condenser, illustrated in section in Fig. 81, represents an improved design supplied by Messrs. Koerting Bros., Ltd., of London. The general principle of construction is similar to that shown in Fig. 77.

The steam inlet branch is placed at the side at A, and the injection water enters at B on the top of the condenser. The cone in the water space ensures an even distribution of the water to the water jets,

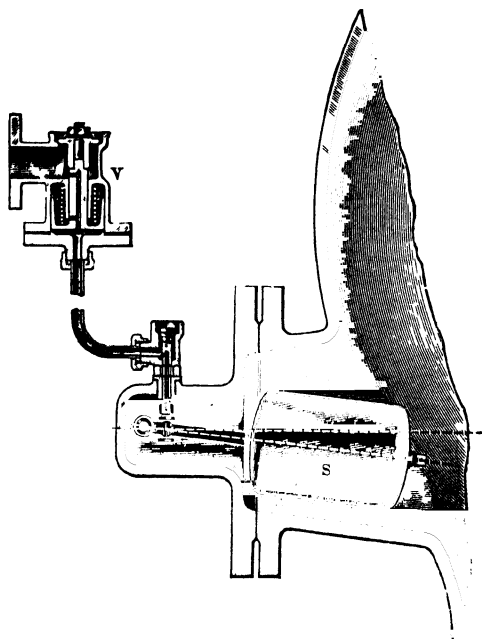


FIG. 82

VACUUM BREAKER FOR KOERTING CONDENSER

and the inverted cone in the centre of the steam space assists the combining of the individual jets into a common jet. The patented automatic vacuum breaker is attached to the main body of the condenser at S. The action of this device is as follows—

The air admission valve V, Fig. 82, is fixed on the exhaust pipe as near as can conveniently be arranged to the engine or turbine. Under normal working conditions the valve is held closed by the atmospheric pressure and the spring which presses on the piston attached to the valve spindle. A channel in the spindle forms a connection between the small chamber above the piston and the

exhaust pipe, whilst a small pipe connects the underside of the piston to the condenser exhaust chamber, so that the pressures on both sides of the piston are practically balanced.

If for any reason water rises in the condenser exhaust chamber, the float S attached to this opens the small relay valve, and air

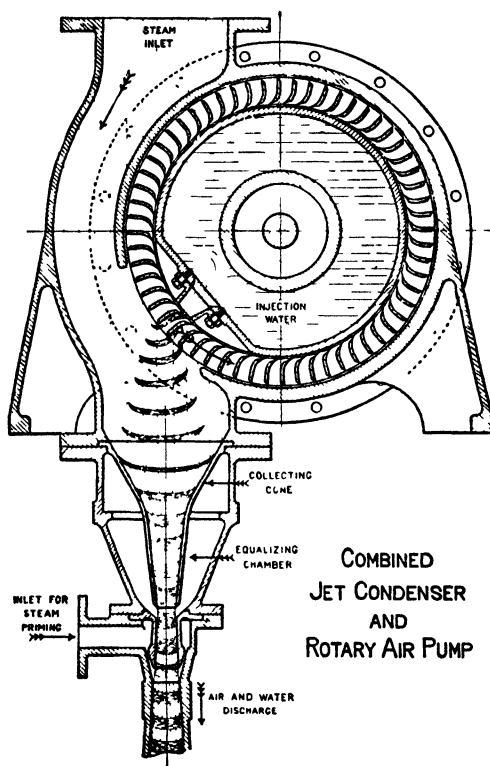


FIG. 83

MIRRIEES-WATSON-LEBIANC ROTARY EJECTOR CONDENSER.

admitted through this presses on the under-side of the piston and, opening the main air valve, breaks the vacuum. The air admitted into the exhaust connections passes to the condenser and clears the water from the exhaust chamber. The float then falls and closes the relay valve, and the pressures on the top and bottom of the piston being again equalized, the air admission valve is automatically closed by the spring.

Mirrlees-Watson Single Jet Condenser. The single jet condenser of Messrs. Mirrlees-Watson's manufacture, illustrated in Fig. 83,

complies with our definition of an ejector condenser in that the condensation of steam and compression of the air take place simultaneously and in a common diffuser. The condenser consists mainly of an air pump of the Mirlces-Leblanc type, described in Chapter XXII. This condenser differs from such air pumps in that the whole of the injection water passes through the pump, and since it is heated by the condensation of the steam it cannot, of course, be passed back to a tank and used over again except after being cooled. The makers recommend the use of this simple form of condenser for duties up to 10,000 lbs. of steam per hour. A steam ejector is usually provided to evacuate the condenser when starting, and so to raise the circulating water to the pump level.

The Rees-Roturbo Rotary Jet Condenser. The condenser (Fig. 84) produced by the Rees-Roturbo Co., Ltd., of Wolverhampton, represents a development of the company's centrifugal pump. It is claimed that this condenser embodies the simplicity of the stationary ejector condenser in its action combined with a low power consumption. The impeller forms the only revolving part, and is so constructed as to perform three distinct functions.

Firstly, the central portion *a*, Fig. 84, or pressure chamber, raises the water through a suction lift up to 18 to 20 ft., and imparts to the water a hydraulic head.

A pair of rings *b* containing a series of nozzles is formed round the periphery of the pressure chamber, and performs the second function of forming a series of water jets that are brought into contact with the incoming steam and condense the steam in the mixing chamber.

Lastly, we have a disc *c* formed between the two nozzle rings, and carrying a series of fan-shaped blades *d* on its periphery that pick up the circulating water together with the condensate and air after condensation has been effected, and eject the mixture through the cutwater *e* into the discharge space *f* against the atmospheric pressure. Since the nozzles revolve with the impeller, the water jets in turn continue to revolve at a comparatively high speed, so that each jet sweeps a large area and a relatively large amount of steam can be condensed in the small mixing chambers.

Whereas in a stationary ejector the kinetic energy that is imparted to the water by the lift pump is utilized for ejecting the mixture from the vacuum space, the fan blades perform this work positively by centrifugal force in the present design. At the same time these blades form an automatic regulator which deals with fluctuating conditions of steam or vacuum without any speed regulation. For instance, when building up the vacuum on starting the condenser,

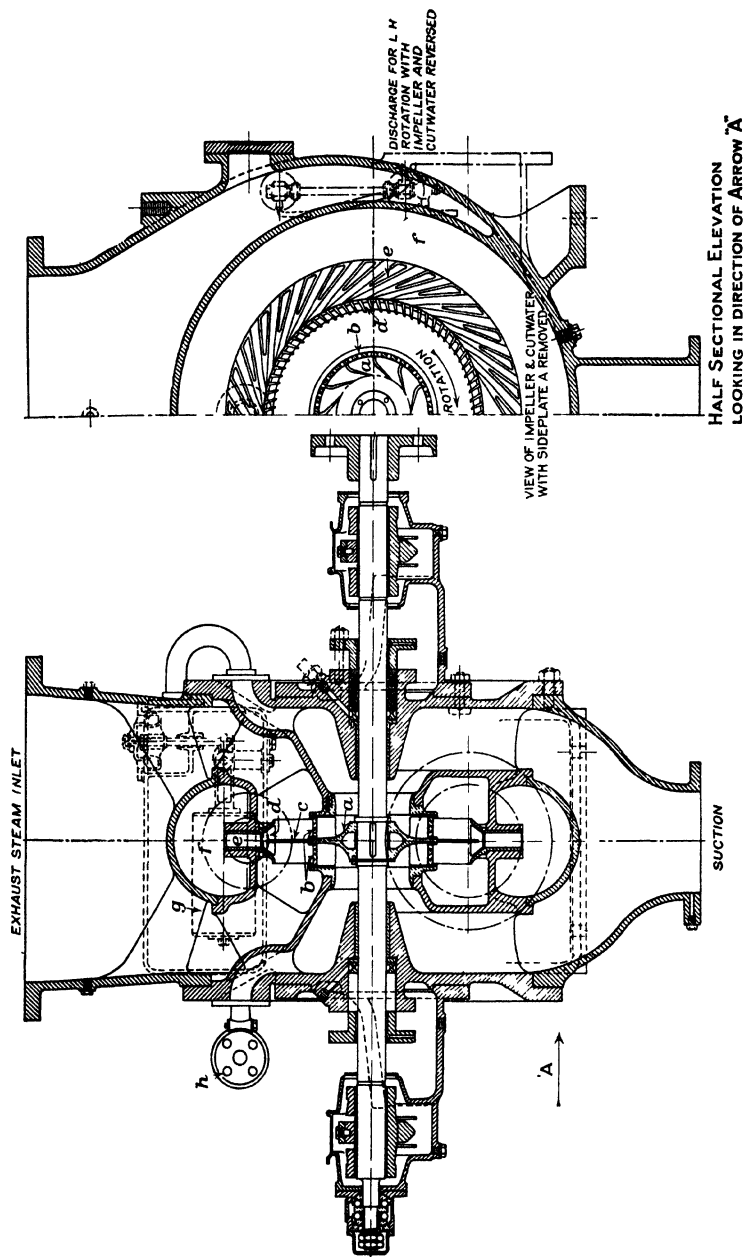


FIG 84.
 REES-ROTURBO ROTARY EJECTOR CONDENSER.

or if the vacuum is reduced below the normal when running, the rim blades are practically empty, and the condensate is swept out through the exhausting fan at full speed. If the vacuum rises beyond the normal, thus exerting a greater backward pull tending to cause "choking," the rim blades fill up with water to a greater depth the higher the vacuum, and because of the extra centrifugal force created by the greater radial depth, and therefore weight, of the water carried round by the fan blades, exercise a greater exhausting effect when running at constant speed.

A vacuum breaker *g* is fitted to admit air in case of failure of the water supply, and the steam ejector *h* is used to prime the condenser.

The makers have standardized six separate sizes of condenser whose rating varies with the specified water temperature and vacuum. The largest size is also supplied as a Duplex unit. The impeller running speeds vary from about 1,440 r.p.m. in the case of the smallest units to 360 r.p.m. for the largest size, and are therefore suitable for direct driving by electric motors or for gear drive by steam turbines. The general design is very compact and well adapted for mounting below a steam turbine, the standard steam inlet flange being so dimensioned as to be suitable for direct bolting to a turbine exhaust flange.

The following figures give particulars of the power and water consumption of Rees-Roturbo condensers, designed for a normal capacity of 20,000 lbs. of steam per hour and working under varying conditions—

	Case 1.	Case 2.	Case 3.
Source of Steam.	Reciprocating Engine.	H.P. Turbine.	H.P. Turbine.
Quantity of steam .	20,000 lbs per hr.	20,000 lbs per hr	20,000 lbs. per hr.
Vacuum .	26 Hg.	28 Hg.	28½ Hg.
Water inlet temp. .	70° F	75° F.	60° F.
Water quantity .	55,000 G.P.H.	98,000 G.P.H	89,000 G.P.H.
B.H.P. .	27	47	42

CHAPTER XVII

AIR PUMPS—RECIPROCATING TYPE CALCULATIONS

IN order to study the effect of an increased inflow of air into a condenser with air extractors of different types, the principle of operation of the extractor must be clearly understood. The Edwards type of pump of which an illustration is given in Fig. 101, will be considered first. In Fig. 85 there is shown a theoretical air pump diagram, actual diagrams being shown in Fig. 86. When the bucket starts its upward stroke the air is compressed along the line CD (Fig. 85) and discharged through the head valves from D to A , AN being the clearance space. The air contained therein is expanded on the return stroke along the line AB . The pressure represented by BC is the amount by which the pressure in the air pump is below the pressure in the condenser.

The laws governing the expansion and compression of gases are two in number—

$$pv = RT, \text{ and}$$

$$pv^\gamma = \text{constant}$$

where p = pressure of gas in lbs. per square foot.

v = volume of 1 lb. of gas in cu. ft., i.e. specific volume.

T = absolute temperature in degrees F.

R = constant and equal to the difference between the specific heats of the gas at constant pressure, and at constant volume measured in ft.-lbs.

γ = ratio of the above specific heats.

For air $R = 53.18$ and $\gamma = 1.408$.

Let $p_1 v_1$ and T_1 be the conditions at A

$$p_2 v_2 \quad ,, \quad T_2 \quad ,, \quad ,, \quad B$$

$$p_3 v_3 \quad ,, \quad T_3 \quad ,, \quad ,, \quad C$$

$$p_4 v_4 \quad ,, \quad T_4 \quad ,, \quad ,, \quad D$$

The above pressures p_1 , p_2 and p_3 refer to the air only.

As explained in Chapter VI, vapours or gases, when mixed in the same space, each exert their normal pressure as if each were present alone, the total pressure at each point being obtained by adding

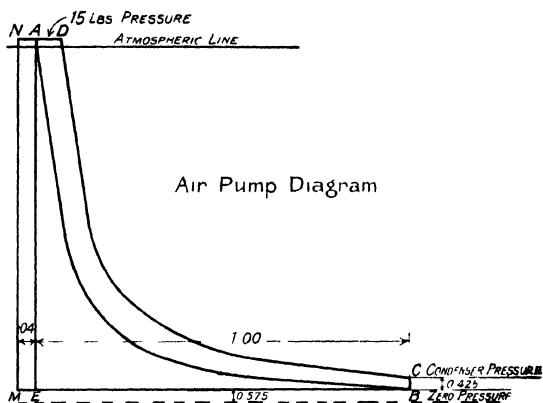


FIG 85
THEORETICAL AIR PUMP DIAGRAM

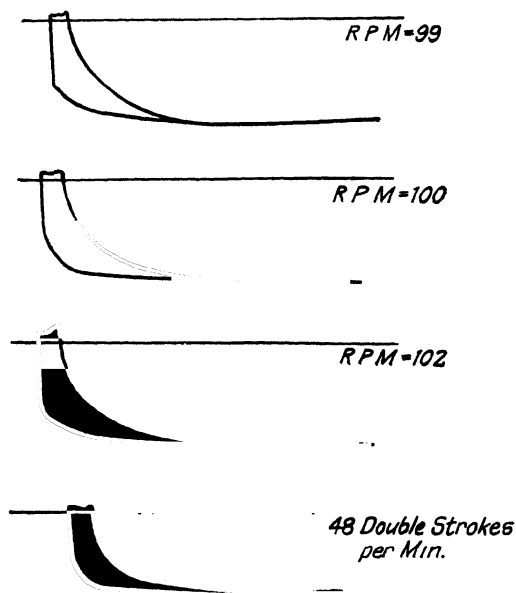


FIG 86
EDWARDS AIR PUMP DIAGRAMS

the partial pressure. In the following brief development of the theory of the action of the reciprocating air pump, attention has been confined to the behaviour of the air, and the contained vapour has been ignored.

Pump Capacity. The expansion of the air along AB , and the compression along CD may be either isothermal or adiabatic.

In air pumps with a separate supply of sealing water or dealing with the condensate as well as the air, the presence of this water ensures that the expansion and compression is isothermal, and this will be dealt with first. By the usual and well-known formula the work done per revolution in compressing the air along CD less the work done by the air in expanding along AB is equal to

$$p_3 v_3 \left(\log_e \frac{p_1}{p_3} + 1 \right) - p_1 v_1 \left(\log_e \frac{v_3}{v_1} + 1 \right) \quad (1)$$

v_3 (or v_2 , which is the same) is the volume swept out by the air pump bucket of the Edwards type, from the top of its stroke down to the point at which the barrel openings are uncovered, plus the clearance space v_1 .

The weight of air removed per stroke of the pump is determined by the difference in pressure between the points B and C , or $(p_3 - p_2)$. Thus, when the ports are uncovered, the whole volume v_2 is filled with air at a pressure p_2 , but at the moment when the ports are covered on the upstroke the same space is filled with air at pressure p_3 .

As the density of air is equal to the reciprocal of the volume per pound or specific volume it is equal to $\frac{p}{RT}$, and therefore the weight of air contained in v_3 at the point B is

$$\frac{p_2 v_3}{RT} \text{ lbs.}$$

Similarly at point C the weight is

$$\frac{p_3 v_3}{RT} \text{ lbs.}$$

The weight of air removed per revolution is less than

$$\frac{(p_3 - p_2) v_3}{RT}$$

as this assumes 100 per cent efficiency. Unless the air pump bucket remained at rest at the bottom of each stroke, and remained there for an appreciable period, the pressure above the bucket cannot

be equal to the condenser pressure. There must always be a pressure difference to induce flow of air into the pump. This pressure difference varies from a maximum when the port is first uncovered to a minimum at the bottom of the stroke. Hence the actual weight removed per revolution is only

$$\frac{\eta(p_3 - p_2)v_3}{RT}$$

where η is the volumetric efficiency of the pump.

As, however, T is constant $p_2 = p_1 \frac{v_1}{v_3}$, and therefore the weight of air removed per revolution is

$$\frac{\eta(p_3 v_3 - p_1 v_1)}{RT} \quad . \quad . \quad . \quad . \quad (2)$$

The revolutions per lb. of air removed are $\frac{RT}{\eta(p_3 v_3 - p_1 v_1)}$ and the work done per lb. of air ignoring all mechanical losses in the pump is

$$\frac{RT}{\eta(p_3 v_3 - p_1 v_1)} \left[p_3 v_3 \left(\log_e \frac{p_1}{p_2} + 1 \right) - p_1 v_1 \left(\log_e \frac{v_2}{v_1} + 1 \right) \right] \quad . \quad (3)$$

Effect of Clearance Space. Obviously, the ratio of the clearance space to the total pump volume limits the lowest pressure which can be obtained by the pump. If v_1 is unduly increased, p_2 is increased proportionately, and the nearer this approaches p_3 the less air is withdrawn by the pump.

If it be assumed that

$$v_3 = n v_1, \text{ and}$$

$$p_3 = m p_1,$$

and these values be substituted in (3), then no work will be done on the air when

$$m \left(\log_e \frac{1}{m} + 1 \right) = \frac{1}{n} (\log_e n + 1)$$

The obvious solution of this is that $m = \frac{1}{n}$, which is also obtainable from equation (1). Thus, assuming that the clearance space is 4 per cent of the effective volume of the pump, then

$$n = \frac{1.04}{.04} = 26.0$$

therefore $m = \frac{1}{26}$, or if $p_1 = 2,016$ lbs. per sq. ft., the minimum pressure in the pump will be $\frac{2016}{26}$ or 77.53 lbs. per sq. ft., about

0.53 lbs. per sq. in., this assuming that $n=100\%$. Thus, for high vacua with reciprocating pumps, a small clearance is needed.

If the expansion and compression in the air pump were adiabatic instead of isothermal, the following relations between the air pressure volume and temperature would hold—

$$\begin{aligned} p_2 &= p_1 \left(\frac{v_1}{v_2} \right)^\gamma \\ p_4 &= p_3 \left(\frac{v_3}{v_4} \right)^\gamma \\ v_4 &= v_2 \left(\frac{p_3}{p_1} \right)^{\frac{1}{\gamma}} \\ T_2 &= T_1 \left(\frac{v_1}{v_2} \right)^{\gamma-1} \\ T_4 &= T_3 \left(\frac{v_3}{v_4} \right)^{\gamma-1} \end{aligned}$$

In the above $p_4 = p_1$ and $v_2 = v_3$. The work done on the air per stroke is given by the expression

$$\frac{\gamma}{\gamma-1} p_1 (v_4 - v_1) - \frac{1}{\gamma-1} v_2 (p_3 - p_2)$$

The weight of air removed per stroke is given by the same formula as for adiabatic expansion and compression, the appropriate values for the pressure and volumes being inserted therein.

Having obtained formulae connecting the pressure and volumes and weight of air removed, the effect of an increase in the inflow of air in the ratio of n to 1 can be noted. The speed of the pump remaining constant, the conditions so alter that a new state of equilibrium is reached when the pump is extracting air at the same rate that it is entering the condenser. But the weight of air removed per revolution was

$$\frac{\eta(p_3 v_3 - p_1 v_1)}{RT}$$

and this must now be increased in the ratio of n to 1. The only variable is p_3 , and so if p_3^1 be the new value

$$p_3^1 v_3 - p_1 v_1 = n(p_3 v_3 - p_1 v_1)$$

and therefore

$$p_3^1 = \frac{n p_3 v_3 - p_1 v_1 (n-1)}{v_3} \quad (5)$$

assuming that η is constant at all pressures, an assumption which is not correct for any considerable change of pressure. (See Fig. 88.)

Thus the partial air pressure in the condenser is increased, but not quite in the same ratio as the inflow of air is increased.

Speeding up the pump would decrease p_3^1 by reducing the air to be removed per revolution.

Variation of Air Pump Efficiency. In a report * published by the North-East Coast Institution of Engineers and Shipbuilders dealing

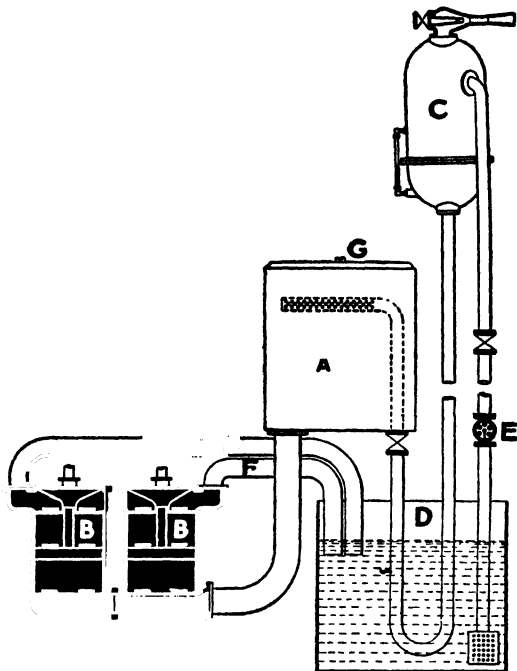


FIG. 87.

DIAGRAMMATIC ARRANGEMENT OF PLANT IN NORTH-EAST COAST TESTS

with certain methods of obtaining vacuum, and prepared by Messrs. Orde, Cairns & Morrow, a large amount of data are given showing the effect of increasing air quantities on the vacuum maintained by reciprocating pumps. In Fig. 87 there is shown diagrammatically the arrangement of the pumps when working as an ordinary wet pump. Air is admitted into the receiver A through a calibrated nozzle G, and exhausted from the receiver A by both barrels B. In order to reproduce the normal working conditions of a condensing plant, condensate is drawn from the

* "On certain methods of producing vacuum," *Proc. N.E. Coast Institute of Engineers and Shipbuilders*, 1916-7.

tank D through the deaerator C and enters the receiver by the spray pipe shown. The water was elevated into C by the steam jet shown, and this jet discharged to atmosphere the greater part of the air contained in the water. It did, at any rate, reduce the air contents to a constant quantity and so rendered possible experiments on a comparable basis.

Each barrel was 20 ins. diameter and 15 ins. stroke. One barrel was direct driven by a steam cylinder on the same line, and

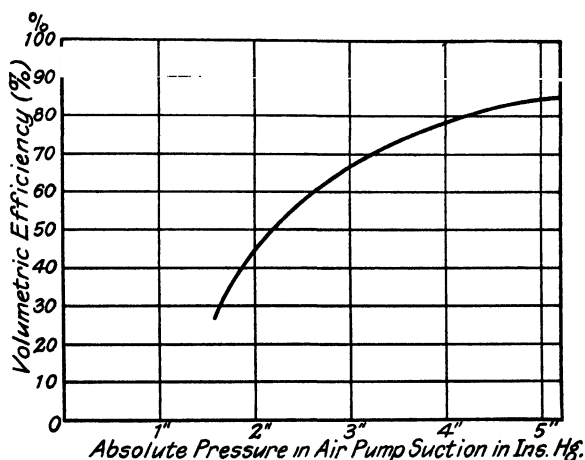


FIG 88

VARIAION OF RECIPROCATING AIR PUMP EFFICIENCY WITH
ABSOLUTE PRESSURE.

the other by means of a beam. The valves were of the Kinghorn type, six being provided as head valves, foot valves, and bucket valves.

The results of Tests No. 2 to 8 are recorded as follows—

No. of Test.	Diameter of Nozzle G.	Absolute Pressure.	Speed of Pump. Revolutions (or Double Stroke) per Minute.
	mm.	Ins. Hg.	
2	2	1.70	39.2
3	3	2.00	39.8
4	4	2.15	39.2
5	5	2.85	39.8
6	6	3.32	39.2
7	7	3.90	39.8
8	8	4.55	39.2

The temperature of the condensate was practically 80° in all tests. The investigators assumed that the air passing through the nozzle G was $0.0244d^2$ lbs. per minute where d = diameter of nozzle in millimetres. This is stated to be correct at 60° F., and no correction factor is introduced for other temperatures.

Dealing with Test No. 2, the theoretical weight of air removed, assuming that the clearance is 4 per cent of the volume swept by the bucket, is 0.007776 lbs. per revolution. The air entering through a 2 mm. nozzle is 0.0976 lbs. per minute or 0.00249 lbs. per revolution at 39.2 r.p.m. This corresponds to an efficiency (volumetric) of 32.02 per cent. The variation of the efficiency with the absolute pressure at the pump suction is shown in the following table and also in Fig. 88.

TABLE IX

No. of Test.	Absolute Pressure, Air Pump Suction.	Air Pump Vol. Efficiency.
	Ins. Hg.	%
2	1.70	32.02
3	2.00	45.82
4	2.15	70.23
5	2.85	63.68
6	3.32	72.90
7	3.90	77.05
8	4.55	82.45

With the exception of Test No. 4, the above efficiencies fall on a fair curve which shows that the air pump efficiency, as might be expected, increases with the density of the air at the pump suction. The greater the condenser pressure, the greater the head, causing the air to flow into the air pump, and the more completely is the air pump barrel filled with air at condenser pressure. The curve in Fig. 88 is characteristic, but the actual value of the efficiency at any absolute pressure will depend upon the design of the pump, speed of rotation, etc.

An increase in steam duty on the condenser may result in a lowered vacuum, but this is a condenser and not an air pump effect except in so far as the air quantity is increased, due to the greater amount of steam entering the condenser. The air pump will maintain a certain partial air pressure in the condenser, constant for a constant rate of air inflow and a constant temperature of the suction, and the total absolute pressure in the condenser will be

the sum of this partial air pressure and the vapour pressure determined by the temperature at the air suction on the condenser.

Desiring to obtain higher vacua with larger generating sets, Sir C. A. Parsons minimized the limitations of the reciprocating pump by using in his vacuum augmentor a steam jet in series with it. This enabled him to increase p_3 at the pump without any corresponding increase in the condenser.

CHAPTER XVIII

AIR PUMPS—STEAM EJECTOR TYPE CALCULATIONS

THE working of a steam jet is simple, but the action consists of a number of separate and distinct operations, each of which has to be considered to arrive at the fundamental equations representing the underlying principles.

In a steam jet there are three masses entering the ejector—

- (1) The operating steam,
- (2) The air to be exhausted,
and
- (3) The vapour contained in the air.

A simple ejector is shown diagrammatically in Fig. 89. The various stages in its operation are as follows—

Steam enters by the jet *J* and, expanding adiabatically, issues from the jet at a high velocity.

Air saturated with vapour enters by the inlet *C* with a velocity appreciable but low by comparison with the steam velocity.

The air and vapour impinge upon the jet of steam and are carried along with it into the diffuser by surface friction. The sum of the momenta of the steam and air before and after impact is constant, and there is a loss of kinetic energy as a result of the impact.

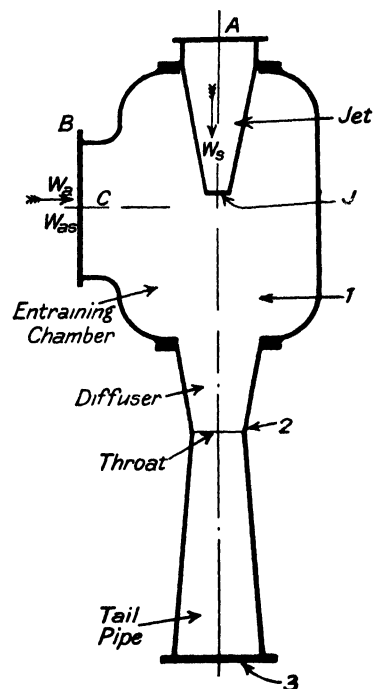


FIG. 89.

DIAGRAMMATIC REPRESENTATION OF
SINGLE STAGE EJECTOR

The kinetic energy thus lost is converted into heat energy, the heat being added to the combined mixture of steam and air. If the steam used in jet *J* is saturated at exit, this increase of heat will raise the

dryness fraction or reduce the supersaturation and may superheat the steam and heat the air, but if superheated or dry saturated at the exit it will raise the temperature of both steam and air. The velocity with which the steam leaves the jet J is affected by the pressure in the ejector, which in turn is assumed to be equal to that in the condenser. The kinetic energy of the steam after mixing with the air is used to compress both steam and air and discharge the combined mixture to the atmosphere. The following calculations are based on the assumption that the pressure in the condenser, in the ejector connection C , and in the ejector down to the entry to the diffuser, are all equal and constant.

The calculations require the use of the following symbols—

A = Area of flow in square feet.

W = Weight flowing in lbs. per second.

h = Heat contents in B.Th.U. per lb.

p = Pressure in lbs. per square foot absolute.

v = Volume of 1 lb. in cubic feet or specific volume.

l = Latent heat of steam in B.Th.U. per lb.

t = Temperature in degrees F.

T = Absolute temperature in degrees F.

V = Velocity in feet per second.

q = Dryness fraction of the steam.

J = Joules mechanical equivalent of heat = 777.8.

To simplify the calculations it is proposed to use in connection with various symbols the suffix, "s" to indicate any property of the steam, "a" any property of the air, and "as" any property of the vapour contained in the air. Thus, as W equals the weight flowing per second, then W_s = steam per second in lbs. W_a = air per second in lbs. W_{as} = vapour in air per second in lbs.

Again, for simplification reasons, additional suffixes will be used to indicate the position in the ejector at which a given property is measured. Thus, assuming that there are positions 1, 2 and 3, then V_{a1} indicates the velocity of air at point 1, V_{s2} the velocity of the steam at point 2, etc. The suffixes 1, 2, 3, c and j will be used to indicate the following positions—

c = Air at inlet from condenser.

j = Outlet of jet.

1 = Entrance to diffuser.

2 = Throat of diffuser.

3 = Outlet of tail pipe.

Thus the total pressure at the air inlet flange C is

$$p_c = \text{air pressure } (p_{ac}) + \text{vapour pressure } (p_{aso}).$$

In calculating certain quantities such as q_2 , intermediate values are obtained which will be indicated by a dash, i.e. q'_2 , etc.

Velocity of Steam at the Jet. Dealing first with the jet, the velocity at exit due to adiabatic expansion in the jet is

$$V_j = 224 \sqrt{\text{heat drop}} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

If the steam has time in its passage through the jet to form moisture, then it should be wet at the exit. But even if the heat drop is only 100 B.Th.U., the velocity is 2,240 ft. per second. The length of the nozzle is usually measured in inches, so that the time of passage is extremely short and is insufficient to allow any portion of the steam to condense. Hence the steam at exit is in a supersaturated condition, i.e. a given volume contains a greater weight of steam than is required to saturate it.

SUPERSATURATED STEAM. Steam in this state behaves as a gas and not as a vapour, and is cooled considerably below the saturation temperature corresponding to its pressure. Just as superheat is measured in the excess of the temperature above saturated temperature, so supersaturation may be expressed by the amount the temperature is below saturation temperature. The value of γ for supersaturated steam is the same as for superheated steam, i.e. 1.308 instead of 1.130, the value for saturated steam.

Callendar has shown that there is a limit to the undercooling due to supersaturation. He initially assumed that on reaching this limit the steam was instantaneously transformed into wet saturated steam in a state of equilibrium. However, this change requires a finite time for its completion so that the supersaturated steam must persist throughout the complete expansion in some degree. Callendar has defined the supersaturation limit as—

“The limit of the state of dry supersaturated steam in rapid expansion which is reached when the pressure is eight times the normal saturation pressure corresponding to the actual temperature of the steam.” (*Properties of Steam*, p. 249.)

Thus, steam expanded down to an absolute pressure of 16 lbs. per sq. in. could be undercooled to the temperature corresponding to a pressure of 2 lbs. or 126.1° F. As the temperature of saturated steam at 16.0 lbs. is 216.3°, this corresponds to an undercooling of 90.2° F.

In the nozzle used in air ejectors, steam is expanded from, say, 150 lbs. dry saturated down to condenser pressure, and will exceed the limit of supersaturation. Under these circumstances the

available heat drop is reduced below that corresponding to adiabatic expansion, the reduction being of the order of 7 to 8 per cent. This will reduce the exit velocity of the steam by approximately 4 per cent. As ejectors are designed with good margins, there is no need to go into great refinements in calculating the effect of supersaturation upon the performance.

DRYNESS FRACTION OF THE STEAM. Using the usual formula equating the entropy before and after expansion in the jet, we can calculate the dryness fraction after expansion, thus—

$$\frac{q_l}{T} + \log_e T = \frac{q'_l}{T_j} + \log_e T_j \quad . \quad . \quad . \quad (2)$$

Due to the efficiency of the nozzle being below unity, i.e. the exit velocity being below the value calculated by equation (1), a certain amount of heat is not converted into kinetic energy but is retained in the steam. This extra heat would be used to dry the steam if it were actually wet, and as the steam is probably supersaturated it reduces the undercooling, raising the temperature nearer to the saturation temperature. Thus, if the coefficient of discharge of the nozzle be θ so that the actual velocity of issue be θV_j , the heat which is used in this manner equals

$$\frac{V_j^2 (1 - \theta^2)}{2gJ} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

This extra heat increases the dryness fraction from q'_l to q_l , so that

$$(q_l - q'_l)l_j = \frac{V_j^2(1 - \theta^2)}{2gJ} \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Action in Entraining Chamber. There is now, entering the entraining chamber, a rapidly moving jet of steam, and in the chamber, assuming the vacuum to be established, saturated air at a low temperature and pressure. If the steam and saturated air were mixed in a vessel in which both were allowed to remain for a sensible period, Dalton's law, according to which the pressure exerted by a mixture of a gas and a vapour, of two vapours or of two gases, is equal to the sum of the pressures which each would exert if it occupied the same space alone, would operate.

But all physical laws require a finite time for their operation, and the time of passage through the entraining chamber is too short for the operation of Dalton's law.

Hence the steam from the jet does not mix with the vapour in the air as it would if at rest, but passes straight through the entraining

chamber, carrying a certain weight of saturated air with it by surface friction. Thus, at the entrance to the diffuser there is—

(1) A central jet of rapidly moving steam in a supersaturated condition, and

(2) An annular surround of saturated air moving at an average velocity below that of the central jet.

As the surface of the central jet is in contact with the more slowly moving saturated air, the surface is moving less rapidly than the steam on the centre line of the jet. Equally as the motion is imparted to the outer annuli of air by friction of the inner annuli, it follows that the outer annuli are moving at a still lower velocity. Thus, at the entrance to the diffuser the velocity is a maximum on the centre line (on the assumption that the motion is entirely rectilinear) falling gradually to the walls of the diffuser. In calculating the average velocity it is usually assumed that the velocity is constant across the whole entrance to the diffuser, and the assumption does not introduce any error of importance compared to other points regarding which it is admitted that the theory is somewhat approximate.

On this assumption equating the momenta before and after mixing, we have

$$(W_a + W_{as}) V_c + W_s V_j \theta = (W_a + W_{as} + W_s) V_1$$

so that $V_1 = \frac{(W_a + W_{as}) V_c + W_s V_j \theta}{W_a + W_{as} + W_s} \quad (5)$

The loss of kinetic energy is

$$\frac{W_a + W_{as}}{2g} V_c^2 + \frac{W_s}{2g} V_j^2 \theta^2 - \frac{(W_a + W_{as} + W_s)}{2g} V_1^2 \quad (6)$$

Conditions at Entry to Diffuser. Dealing with the volumes at the entrance to the diffuser we have a central jet consisting of steam only (ignoring the air carried from the boiler in the jet steam) at pressure p_j and temperature t_j , moving at a velocity V_1 . The weight flowing per unit of time is W_s and the dryness fraction is q_1 . If v_{s1} be the specific volume in cubic feet per lb., the total volume passing per unit of time is

$$W_s v_{s1} q_1$$

and the area required to accommodate this jet is

$$\frac{W_s v_{s1} q_1}{V_1} \quad (7)$$

by the total weight of steam present, we get the specific volume of the mixture, thus

$$W_s q_{s1} v_{s1} + W_{as} v'_{as1} = (W_s q_{s1} + W_{as}) v_{s1} \quad (11)$$

in which $v'_{as1} = v_{a1} \frac{W_a}{W_{as}}$ by Dalton's law.

From the steam tables, or more conveniently from a steam chart, knowing the heat contents and specific volume, the pressure p_{s1} can be obtained.

The pressure and volume were previously p'_{a1} and v'_{a1} and the temperature T'_{a1} . The new volume is v_{a1} and the temperature T_1 corresponding to p_{s1} , so that

$$p_{a1} = p'_{a1} \frac{v'_{a1}}{v_{a1}} \cdot \frac{T_1}{T'_{a1}} \quad (12)$$

The total volume occupied by the air is the same as the total volume of the steam. Hence

$$(W_s q_{s1} + W_{as}) v_{s1} = W_a v_{a1}$$

$$\text{or } v_{a1} = v_{s1} \frac{W_s q_{s1} + W_{as}}{W_a}$$

Inserting this in equation (12) we get

$$p_{a1} = p'_{a1} \frac{v'_{a1}}{v_{s1}} \cdot \frac{W_a}{W_s q_{s1} + W_{as}} \cdot \frac{T_1}{T'_{a1}} \quad (13)$$

Both air and steam partial pressures at the entrance to the diffuser are now known.

In the diffuser the mixed air and steam are to be compressed by the conversion of kinetic energy into pressure energy.

Conversion of Kinetic to Pressure Energy. The connection between the pressure and velocity of any gas in motion is determined from the gas equation

$$\frac{V^2}{2g} + \int \frac{dp}{w} = \text{constant}$$

which assumes that there are no losses in the process. As $p v^\gamma = \text{constant} = p_1 v_1^\gamma$

$$\frac{1}{w} = v = v_1 \left(\frac{p_1}{p} \right)^{\frac{1}{\gamma}}$$

and therefore

$$\frac{V^2}{2g} + p_1^{\frac{1}{\gamma}} v_1 \int \frac{dp}{p^{\frac{1}{\gamma}}} = \text{constant}.$$

From this

$$\frac{V^2}{2g} + \frac{\gamma}{\gamma-1} p_1^{\frac{1}{\gamma}} v_1 p^{\frac{\gamma-1}{\gamma}} = \text{constant.}$$

When $p = p_1$, $V = V_1$, and therefore

$$\text{Constant} = \frac{V_1^2}{2g} + \frac{\gamma}{\gamma-1} p_1 v_1$$

So that, if we assume that $\frac{p}{p_1} = m$

$$\text{then } V^2 = V_1^2 - 2g p_1 v_1 \frac{\gamma}{\gamma-1} (m^{\frac{\gamma}{\gamma-1}} - 1) \quad . \quad . \quad (14)$$

The actual velocity at any point is less than the value given by this equation on account of the inefficiency of the diffuser.

Adiabatic Compression of a Mixture. Equation (14) would give, by using the correct values for γ , the connection between velocity and pressure for either air or steam separately, but in the diffuser they are intimately mixed and the correct value for γ is intermediate between the air and steam values. The exact value may be determined in the following manner.

It is shown on page 195 that in any change of condition of a gas, the heat change dH can be completely determined if the change in two of the three variables, pressure dp , temperature dt , or volume dv are known, and that

$$dH = \frac{1}{\gamma-1} d(pv) + p dv$$

If γ_a and γ_s be the exponents for the adiabatic compression of air and steam respectively, and dH_a and dH_s , the heat changes, then

$$dH_a = \frac{1}{\gamma_a-1} d(p_a v_a) + p_a dv$$

and

$$dH_s = \frac{1}{\gamma_s-1} d(p_s v_s) + p_s dv$$

Integrating we get

$$H_a = \frac{p_a v_a}{\gamma_a-1} + \text{work done}$$

and

$$H_s = \frac{p_s v_s}{\gamma_s-1} + \text{work done.}$$

If γ be the adiabatic exponent for the mixture, then

$$H = \frac{pv}{\gamma - 1} + \text{work done.}$$

But $H = H_a + H_s$, provided that there is no chemical action between the air and steam, an assumption which does not hold for all mixtures of gases, so that

$$\frac{pv}{\gamma - 1} = \frac{p_a v_a}{\gamma_a - 1} + \frac{p_s v_s}{\gamma_s - 1}$$

In accordance with Dalton's law

$$v = v_a = v_s$$

and therefore

$$\frac{p}{\gamma - 1} = \frac{p_a}{\gamma_a - 1} + \frac{p_s}{\gamma_s - 1} \quad (15)$$

This gives the required adiabatic exponent for any mixture of air and steam if the two partial pressures are known. If $p_a = 0$ then $\gamma = \gamma_s$ and vice versa. $\gamma_a = 1.408$ and $\gamma_s = 1.130$.

At any section of the diffuser with area A the volume flowing per second is VA , and the weight flowing through the area A is $\frac{VA}{v}$ lbs. per second. Substituting for V the value from equation (14) and for v_1 the expression $\left(\frac{p_1}{p}\right)^{\frac{1}{\gamma}} v_1$ or $v_1 m^{\frac{1}{\gamma}}$ we have

$$W = Am^{\frac{1}{\gamma}} \sqrt{\frac{V_1^2}{v_1^2} - \frac{2g\gamma}{\gamma - 1} \cdot \frac{p_1}{v_1} \left(m^{\frac{\gamma}{\gamma - 1}} - 1\right)} \quad (16)$$

As the mass flow through the diffuser or W is constant, it follows that A varies inversely as the remainder of the right-hand side of equation (16).

The area is a minimum at the throat of the diffuser, and the pressure there can be obtained thus—

$$\text{Let } x = \frac{V_1^2}{v_1^2} m^{\frac{2}{\gamma}} - \frac{2g\gamma}{\gamma - 1} \frac{p_1}{v_1} m^{\frac{2}{\gamma}} \left(m^{\frac{\gamma}{\gamma - 1}} - 1\right)$$

then A is a minimum when x is a maximum or when $\frac{dx}{dm} = 0$.

When $\frac{dx}{dm} = 0$, then

$$m = \left(\frac{\frac{2gp_1}{\gamma - 1} + \frac{1}{\gamma} \frac{V_1^2}{v_1}}{\frac{\gamma + 1}{\gamma - 1} gp_1} \right)^{\frac{\gamma}{\gamma - 1}} \quad (17)$$

As a matter of interest it may be noted that if $V_1 = 0$ as in the case of a steam nozzle, the above equation reduces to

$$m = \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}}$$

which is a well-known expression.

If the above value for m_1 , giving the ratio of the throat pressure to p_1 , be substituted in equation (14), we get after simplifying

$$V_2'^2 = \frac{\gamma - 1}{\gamma + 1} \left(V_1^2 + \frac{\gamma}{\gamma - 1} 2g p_1 v_1 \right) \quad (18)$$

It should be noted that the value of m at the throat, and also the velocity at the same place, are only dependent upon the conditions existing at the entrance to the diffuser, p_1 , v_1 and V_1 . Thus, the exit or atmospheric conditions do not affect the pressure at the throat.

In order that the throat pressure p_2 may be raised to the inter-cooler pressure by a further drop in the velocity, a divergent section must be added to the diffuser beyond the throat. Equation (16) gives the relation between the pressure and velocity at any point in the diffuser before or after the throat.

Determining Area of Diffuser. In order to determine the area of the diffuser at various points, the partial pressures and specific volumes of the steam and the dryness fraction of the steam must be ascertained.

Dealing with the throat area in particular it will be assumed that the steam remains saturated throughout the compression. The value of γ is obtained from equation (15). Substituting this value in equation (17) we get the value for m at the throat, and so ascertain the steam pressure there p_{s2} . From the steam tables, T_2 , l_2 and v_{s2} can be obtained. The actual volume per lb. of steam is dependent upon the dryness fraction q_2 , which can be determined from equation (2) with suitable substitutions.

The above calculations have been made upon the basis of adiabatic expansion, but the loss of velocity, due to friction and inefficiency of the diffuser, increases the heat contents of the steam, thus raising the dryness fraction or adding superheat to the steam.

It will be assumed that the steam remains saturated throughout the compression and, further, that the efficiency of the diffuser is such that

$$\frac{\text{Energy used to compress steam and air mixture}}{\text{Kinetic energy taken from the moving mixture}} = \psi^2$$

so that total kinetic energy abstracted

$$= \frac{\text{Work done}}{\psi^2}$$

and the extra energy abstracted which is converted into heat is equal to

$$\text{Work done} \left(\frac{1}{\psi^2} - 1 \right)$$

or expressed in heat units

$$= \frac{\text{Work done} (1 - \psi^2)}{\psi^2 J}$$

The dryness fraction was q_2' and is now raised to q_2 , so that

$$(q_2 - q_2') l_2 = \frac{\text{work done} (1 - \psi^2)}{\psi^2 J} \quad . \quad . \quad . \quad (19)$$

As the mixture is assumed to follow the law $p v^\gamma$ during the compression, the work done per lb. of mixture is given by

$$\frac{p_1 v_1}{\gamma - 1} \left(1 - \left(\frac{p}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} \right)$$

where γ has the value obtained from equation (15). If this be inserted in equation (19) the increase in q_2 due to heat added can be calculated. Hence, the actual volume per lb. of steam is $q_2 v_{s2}$, and the total volume at the throat $(W_s + W_a) q_2 v_{s2}$.

In order to calculate the value of V_2 , taking into account the kinetic energy transformed into heat energy, we have, as energy varies as the square of the velocities,

$$\frac{V_1^2 - V_2'^2}{V_1^2 - V_2^2} = \psi^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

As V_1 and V_2' are known, and ψ^2 has an assumed value of, say, 0.85, V_2 can be calculated. In assuming an arbitrary value of 0.85 for ψ^2 , it should be remembered that the efficiency of the diffuser must be considerably lower than that of a nozzle.

Should the conditions be such that the steam becomes superheated during compression in the diffuser, it is more convenient to use a Callendar steam diagram than to evolve and use mathematical expressions.

Use of Callendar Steam Diagram. In Fig. 90 there is shown in diagrammatic form a Callendar steam diagram. Lines A are lines of constant quality or dryness fraction. Line B is the saturation curve, and lines C, lines of constant temperature. Lines D are lines of constant pressure in both the saturated and superheated

regions, and lines E of constant entropy. Assume that point M represents the condition of the steam at the inlet to the diffuser, i.e. $p_1 q_1$. Adiabatic compression of the steam alone is shown by the line MN, parallel to lines E, and terminating on a line representing the terminal pressure. Being above line B the steam at N is already superheated. The added heat is $\frac{\text{work done } (1 - \psi^2)}{\psi^2 J}$

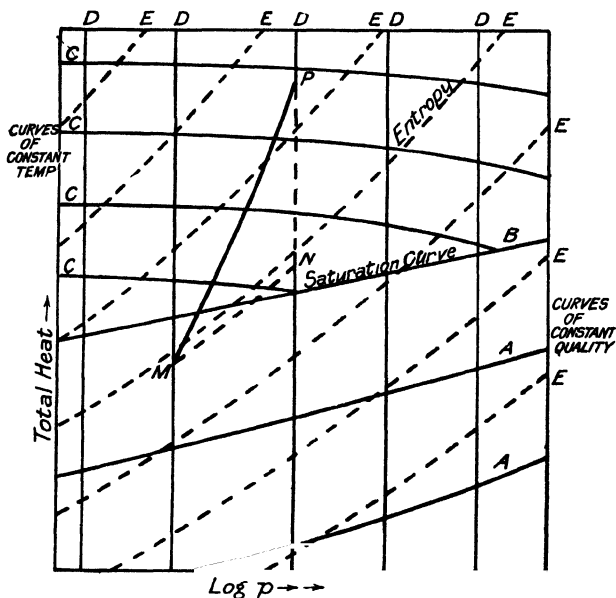


FIG. 90.

TYPICAL CALLENDAR HEAT DIAGRAM.

and the state of the steam is indicated by a point on the same pressure line but farther into the superheated region, such as the point P, so that NP is equal to the added heat units. Thus, the change of state in the diffuser is represented by the line MP.

The specific volume for the superheated steam corresponding to point P in Fig. 90 can conveniently be obtained from the same Callendar steam diagram, which has lines of constant specific volume marked thereon. Thus, $T_2 p_{s2}$ and v_{s2} are known, and the area at the throat is calculated from

$$A_2 = \frac{(W_s + W_{as}) q_2 v_{s2}}{V_2} \quad (21)$$

Ejector with Water Jacket Diffuser. A form of steam air ejector has been introduced in which the diffuser is water-jacketed, and this requires separate mathematical treatment. In the ordinary diffuser as considered previously the compression is approximately adiabatic, and therefore the temperature of the air at the discharge point is raised. Should the conditions be such that the steam becomes superheated the temperature is raised still further. In a water-jacketed diffuser the compression is isothermal, the cooling water removing the heat due to compression and also the extra heat due to the inefficiency of the diffuser. The volume of the air is therefore considerably reduced as compared with adiabatic compression.

The relation between velocity of the air and steam and the pressures is obtained by substituting in the gas equation for $\frac{1}{w}$ the expression $v = \frac{p_1 v_1}{p}$. Differentiating as before we get

$$V^2 = V_1^2 - 2gp_1v_1 \log_e \frac{p_1}{p} \quad . \quad . \quad . \quad (22)$$

As before, the weight of gas passing (W) is equal to

$$\frac{AV}{v} \text{ or } W = \frac{pA}{p_1 v_1} \sqrt{V_1^2 - 2gp_1v_1 \log_e \frac{p_1}{p}} \quad . \quad . \quad (23)$$

In this equation Wp_1 and v_1 are constant, and therefore A is a minimum when

$$p \sqrt{V_1^2 - 2gp_1v_1 \log_e \frac{p_1}{p}}$$

is a maximum. Squaring this and equating it to x we have $\frac{dx}{dp} = 0$, when

$$\log_e \frac{p_2}{p_1} = \frac{V_1^2 - gp_1v_1}{2gp_1v_1} \quad . \quad . \quad . \quad (24)$$

the suffix 2 indicating that we are now dealing with the throat conditions.

Substituting this value in equation (22) we get

$$V_2 = \sqrt{gp_1v_1} \quad . \quad . \quad . \quad . \quad (25)$$

As was the case with adiabatic compressions, the velocity V_2 and pressure p_2 at the throat are dependent upon the initial condition only and not upon the conditions at the exit from the diffuser. It should be further noted that V_2 is independent of V_1 , and only dependent upon the pressure and specific volume at the entry to the

diffuser. V_2 or the throat velocity is equal to the isothermal velocity of sound in gas at the inlet pressure and specific volume.

If it is necessary to use circulating water for the cooling of the convergent diffuser, the heat abstracted to keep the action isothermal is lost to the system, and this should be considered when comparing the two systems.

In any gas whose change of state is governed by the equation

$$\frac{pv}{RT} = \text{constant}$$

the complete condition is defined when any two quantities out of the three p , v and T are known. In the following, t will be used instead of T for convenience. In any change, assume that dH is the change of heat corresponding to any change in the other conditions, then the following equations are obtainable—

$$dH = kdt + ldv \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

[illegible]

[illegible]

where k, l, K, L, P and V are constants.

Equating (26) and (27) and substituting

$$dp = \left(\frac{dp}{dt}\right) dt + \left(\frac{dp}{dv}\right) dv$$

$$kdt + ldv = Kdt + L \left(\frac{dp}{dt} \right) dt + L \left(\frac{dp}{dv} \right) dv$$

Assuming that $dt = 0$ and dividing by dv

$$l = L \left(\frac{dp}{dv} \right) \\ = -L \frac{RT}{v^2}$$

Now assuming that $dv = 0$ and dividing by dt

$$k = K + L \left(\frac{dp}{dt} \right)$$

$$\text{but } \frac{dp}{dt} = \frac{R}{\eta} \text{ and } (K - k) = R$$

so that $K - k = -L \frac{R}{n}$

or $L = -v$

$$\text{Hence } l = \frac{RT}{v} = p$$

$$\text{Similarly } V = p + \frac{kt}{v}$$

$$\text{and } P = \frac{kv}{R}$$

Substituting in equation (28)

$$dH = \frac{kv}{R} dp + \left(p + \frac{kt}{v} \right) dv$$

$$\text{but } \frac{t}{v} = \frac{p}{R} = \frac{p}{K-k}$$

$$\begin{aligned} \text{and } dH &= \frac{k}{K-k} v dp + \frac{kp}{K-k} dv + p dv \\ &= \frac{1}{\gamma-1} [v dp + p dv] + p dv \\ &= \frac{1}{\gamma-1} d(pv) + p dv \end{aligned}$$

If the compression follows the equation

$$p v^s = c$$

$$\text{then } H = \frac{1}{\gamma-1} [pv] + \frac{pv}{1-s} + \text{constant}$$

If $H = 0$ when conditions are p_o, v_o , then

$$\text{constant} = -\frac{1}{\gamma-1} (p_o v_o) - \frac{p_o v_o}{1-s}$$

$$\text{and } H = \frac{pv - p_o v_o}{\gamma-1} + \frac{pv - p_o v_o}{1-s}$$

$$H = \frac{pv - p_o v_o}{s-1} \left(\frac{\gamma-s}{\gamma-1} \right)$$

$$\text{but } \frac{pv - p_o v_o}{s-1} = \text{work done in compression}$$

$$\therefore H = \frac{\gamma-s}{\gamma-1} \times \text{work done.}$$

With isothermal compression $s = 1$, and so the heat rejected to the water is equal to the work done. It may be noted as a matter

of interest that with adiabatic compression $S = \gamma$, and so $H = 0$ corresponding to no transfer of heat.

Conditions on Establishing a Vacuum. All the above calculations are based upon the assumption that the vacuum is established, and that the only duty of the ejector is to maintain it unchanged. But the conditions when starting are very different to those when the vacuum is established. To be acceptable as a practical air ejector it is essential that it shall be able to build up a vacuum in a reasonable time.

When starting up a condensing plant the entraining chamber below the steam jet is filled with air at atmospheric pressure, and therefore the heat drop available is only approximately half as great as when the vacuum is established. The velocity with which the steam issues is therefore reduced by about 30 per cent. In order to gain a clear comprehension of the action, consider for the moment that the steam is not extracting air, although working in air at atmospheric pressure. The steam would issue from the jet, and having no external work to perform it would flow across to and through the diffuser with undiminished velocity.

The throat of the diffuser would have to be the same area as the outlet of the jet. Now imagine the steam to entrain some air. Its velocity is reduced and the cross-sectional area of the central jet of steam is increased. In addition, a certain volume of air is moving with the jet and area must be provided for this. Thus the extraction of air increases the area required at the entrance to the diffuser and at all points therein. If at any point the area is insufficient, the jet of steam will come in contact with the diffuser and prevent the air being exhausted. The air will then eddy in the entraining chamber and a portion of the diffuser, but the vacuum will not build up. With sufficient diffuser area to allow the air to be exhausted, the velocity of the steam issuing from the jet is increased with corresponding reduction in the sectional area of the jet of steam and air. In addition to this reduction, the rate at which the air is exhausted will be reduced continually as the air originally present in the condenser is gradually reduced. Thus the area required for the extraction of the air is reduced both on account of the increased velocity and the reduced quantity. When a certain vacuum is reached the sectional area of the stream of steam and air may be so reduced by comparison with the sectional area under starting conditions, that air may leak back through the throat and prevent the vacuum being increased.

Fig. 91 shows diagrammatically the conditions with insufficient throat area. The steam is in contact with the diffuser and the air

cannot get through. In Fig. 92 the area is correct and the vacuum is established, but in Fig. 93 the maximum vacuum has been reached as air is leaking back round the emerging jet of air and steam.

This shows that a large diffuser area is required for starting conditions and a much smaller area for normal working conditions. Adjustable diffusers have been tried but have not survived the

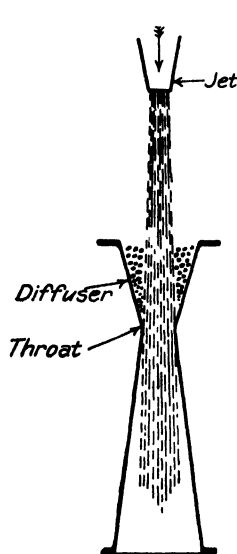


FIG. 91.
Throat Small.

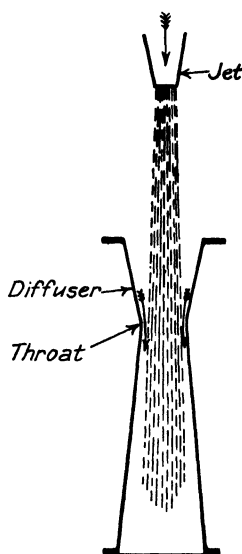


FIG. 92.
Throat Correct.

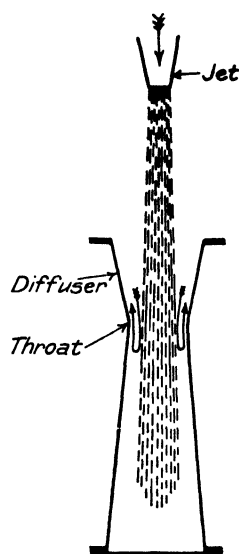


FIG. 93.
Throat Too Large.

STARTING CONDITIONS IN AN EJECTOR.

experimental stage. To these some reference is made on pages 241-2. It is, therefore, necessary to limit the ratio of compression in a single ejector, and practice shows that the ratio should not exceed about seven. With a 29 in. vacuum the total ratio required is about 30, and this would be divided between two ejectors in series with an average ratio of about 5.5.

Calculating the Proportions of a Steam Jet. The calculation of the proportions of a steam jet ejector for any particular set of conditions is a long and tedious process, but an example will be given, as a better understanding of any problem is usually obtained from concrete examples.

In working out an example it will be necessary to work out the weight of vapour to be removed per lb. of dry air. In the following

table* there is given for reference purposes the ratio of the total weight of air and vapour to be removed to the weight of the air alone.

TABLE X

TOTAL WEIGHT OF AIR AND VAPOUR IN LBS TO BE REMOVED BY AIR PUMP
AT VARIOUS TEMPERATURES AND ABSOLUTE PRESSURES PER LB
OF DRY AIR

Absolute Pressures. (Ins Hg.)	Air Pump Suction Temperatures										
	100°	95°	90°	85°	80°	75°	70°	65°	60°	55°	50°
0.9							3.90	2.41	1.87	1.59	1.43
1.0						5.48	2.78	2.03	1.68	1.48	1.36
1.1						3.46	2.29	1.82	1.57	1.41	1.31
1.2					4.88	2.70	2.10	1.68	1.48	1.36	1.28
1.25					3.98	2.47	1.91	1.62	1.45	1.34	1.26
1.3					3.42	2.30	1.83	1.58	1.42	1.32	1.25
1.4				6.66	2.76	2.05	1.72	1.50	1.37	1.28	1.22
1.5				3.67	2.38	1.88	1.61	1.44	1.33	1.26	1.20
1.6			6.04	2.48	2.14	1.76	1.54	1.40	1.30	1.23	1.19
1.7			4.22	2.57	1.97	1.67	1.48	1.36	1.28	1.21	1.17
1.75			3.72	2.42	1.90	1.63	1.46	1.35	1.27	1.20	1.17
1.8		8.52	3.36	2.30	1.84	1.59	1.44	1.33	1.26	1.20	1.16
1.9		5.36	2.86	2.11	1.74	1.54	1.40	1.30	1.24	1.19	1.15
2.0		4.07	2.54	1.97	1.67	1.49	1.37	1.28	1.22	1.17	1.14
2.1	8.44	3.37	2.31	1.86	1.60	1.45	1.34	1.26	1.20	1.16	1.13
2.2	5.65	2.93	2.14	1.77	1.56	1.41	1.32	1.25	1.19	1.15	1.12
2.25	4.81	2.73	2.07	1.73	1.53	1.40	1.31	1.24	1.19	1.15	1.12
2.3	4.29	2.63	2.01	1.70	1.51	1.38	1.30	1.23	1.18	1.15	1.12
2.4	3.59	2.41	1.93	1.64	1.47	1.36	1.28	1.22	1.17	1.14	1.11
2.5	3.13	2.24	1.83	1.59	1.44	1.34	1.26	1.21	1.16	1.13	1.11
2.6	2.81	2.11	1.76	1.55	1.41	1.32	1.25	1.20	1.16	1.13	1.10
2.7	2.57	2.00	1.70	1.51	1.39	1.30	1.24	1.19	1.15	1.12	1.10
2.75	2.48	1.95	1.67	1.49	1.38	1.29	1.23	1.18	1.15	1.12	1.10
2.8	2.39	1.91	1.65	1.48	1.36	1.28	1.22	1.18	1.14	1.12	1.09
2.9	2.26	1.84	1.60	1.45	1.35	1.27	1.21	1.17	1.14	1.11	1.09
3.0	2.13	1.77	1.56	1.43	1.33	1.26	1.20	1.16	1.13	1.11	1.09
3.5	1.77	1.56	1.44	1.33	1.26	1.21	1.17	1.14	1.11	1.09	1.07
4.0	1.58	1.44	1.34	1.29	1.22	1.17	1.14	1.12	1.09	1.08	1.06
4.5	1.47	1.36	1.29	1.23	1.19	1.15	1.12	1.10	1.08	1.07	1.06
5.0	1.39	1.31	1.25	1.20	1.16	1.13	1.11	1.09	1.07	1.06	1.05

We have shown in Chapter VI that the temperature of the mixture of air and vapour at the air pump suction depends primarily on the vapour quantity that can be carried over with the air. In average conditions it may be taken that an air pump suction temperature, corresponding to a partial air pressure of 10 per cent and a partial steam pressure of 90 per cent, will give a reasonably low vapour

* See *Beama*, December, 1920, "Air in Condensers," by I. V. Robinson.

quantity coupled with an amount of air cooling in the condenser that is not excessive.

The figures given in Table XI show the relation between partial air and vapour pressures for a total absolute pressure of 1 in. or 29 in. vacuum.

TABLE XI

Suction Temperature.	Vapour Pressure.	Air Pressure.
78.9°	1.000 in.	0.000 in.
75.77°	0.900 in.	0.100 in.
75°	0.878 in.	0.122 in.
70°	0.741 in.	0.259 in.
65°	0.624 in.	0.376 in.
60°	0.523 in.	0.477 in.
55°	0.437 in.	0.563 in.
50°	0.364 in.	0.636 in.

The figures given below show the temperature at the air pump suction for $p_{ar} = 10$ per cent of p_v for representative vacua.

TABLE XII

Vacuum at Air Pump Suction.	Temperature at Air Pump Suction.
29 in.	75.77°
28 in.	97.61°
27 in.	111.35°
26 in.	121.56°

At each vacuum, provided that the partial air pressure is 10 per cent of the total absolute pressure at the air pump suction, the weight of vapour removed per lb. of air is about 5.62 lbs., or the total weight of mixture to be handled per lb. of air is 6.62 lbs.

When dealing with the condensation of steam which contains air (pages 34-37) the question of air cooling in relation to the amount of vapour removed with the air was considered. With the above assumption in partial air pressure and, say, x lbs. of air per 10,000 lbs., the percentage of vapour removed is

$$\frac{5.62 \times 100}{10,000} \text{ or } 0.0562 x \text{ per cent.}$$

The maximum value of x should not be greater than 5, so that the vapour removed should not exceed 0.28 per cent.

The conditions assumed will be

Vacuum at air pump suction	28.5 in. (bar. 30 in.).
Steam supplied to ejector	660 lbs. per hour.
Air to be withdrawn	38 lbs. per hour (corresponding to about a 5,000 kw. unit).
Temperature of air suction on condenser	88.2° F.
Steam pressure at ejector	200 lbs. per sq. in. pressure, dry saturated.

The ejector will be of the two-stage type and will be assumed to be of the intercooler type, as otherwise the steam consumption would be considerably higher than stated. It will be further assumed that one-third of the steam is used in the first stage.

Then the assumed data are—

$$\begin{aligned}
 W_s &= 0.0612 \text{ lbs. per second.} & p_j &= 0.735 \times 144 \text{ lbs. per sq. ft.} \\
 W_a &= 0.0105 \text{ „ „ „} & t_j &= 91.5^\circ \\
 t_c &= 88.2^\circ & T_j &= 551.1^\circ \\
 T_c &= 547.8^\circ
 \end{aligned}$$

Using Callendar's Steam Tables (Abridged Edition, Fahrenheit Units)* the following additional data can be obtained—

$$\begin{aligned}
 p_{asc} &= 0.661 \times 144 \text{ lbs. per sq. ft.} \\
 \text{and } \therefore p_{ac} &= p_j - p_{asc} = 0.0735 \times 144 \text{ lbs. per sq. ft.} \\
 v_{asc} &= 492.5 \text{ cu. ft. per lb.} \\
 v_{ac} &= \frac{RT}{p_{ac}} = \frac{53.18 \times 547.8}{0.0735 \times 144} = 2.752 \text{ cu. ft. per lb.} \\
 v_{sj} &= 445.5 \text{ cu. ft. per lb.} \\
 h_j &= 1097.7 \text{ B.Th.U. per lb.} \\
 l_j &= 1038.4 \text{ „ „} \\
 h_c &= 1096.1 \text{ „ „}
 \end{aligned}$$

In abstracting the above data the effect of supersaturation has been ignored for reasons indicated on page 185.

Hence vapour per lb. of air = $\frac{2752}{492.5} = 5.58$ lbs., so $w_{as} = w_a \times 5.58 = 0.0105 \times 5.58 = 0.0586$ lbs. per second. From the heat drop tables published for the Turbine Section of the British Electrical and Allied Manufacturers' Association,* and based upon Callendar's steam tables, the heat drop available in adiabatic expansion from 200 lbs. per sq. in. absolute dry saturated to a vacuum of 28.5 ins. is 356.7 B.Th.U. From equation (1)

$$V_j = 224 \sqrt{356.7} = 4,230 \text{ ft. per second.}$$

Assume that the coefficient of discharge of the jet θ is 0.95, which value is probably on the low side.

* By Edward Arnold & Co., Ltd., London.

During expansion the steam becomes wet, the dryness fraction being determined by (2). If steam is initially dry saturated, then $q_j' = 0.7602$. Due to the loss of velocity the heat added to the steam is $\frac{V_j^2(1-\theta^2)}{2gJ}$ or 34.82 B.Th.U. per lb. of steam. Inserting in equation (4) this increases the dryness fraction by .03355 making $q_j = 0.7938$ or, say, $q_j = 0.800$. Thus, of the 0.0612 lbs. steam per second in the jet, approximately 20 per cent or 0.0122 lbs. is present as water, and the total weight of steam present is about 0.1078 lbs. per second.

Applying equation (5) assuming V_c to be 50 ft. per second (but V_c will be found to be negligible)

$$V_1 = \frac{(0.0105 + .05686) 50 + (.0612 \times 4230 \times 0.95)}{.0105 + .0586 + .0612}$$

$$= 1,888 \text{ ft. per second.}$$

From equation (6)

$$\text{Loss of kinetic energy} = 8,139 \text{ ft.-lbs. per lb.}$$

This energy which is equivalent to 10.47 B.Th.U. per lb. of steam is used to dry the centre jet of steam by evaporating some of the moisture present. The increase in dryness fraction is equal to

$$\frac{10.47}{1038.4} = .01007$$

This added to the previous value, makes the value of q_1 0.8039.

Dealing with the mixing, which occurs in the diffuser of the vapour in the air and the steam we get from equation (10)

$$h_1 = 1096.8$$

and from equation (11)

$$v_{s1} = 471.1$$

From the Callendar steam chart with these values of h_1 and s_1 we get $p_{s1} = 0.693 \times 144$ lbs. per sq. ft., and the volume of the steam 0.1078×471.1 cu. ft. = 50.78 cu. ft. The air occupies the same volume, and we shall assume that it is raised to the temperature corresponding to $p_{s1} = 0.693$ lbs. or 89.5° F. This would cause a small amount of condensation, but the amount is negligible.

Hence, the air pressure p_{a1} given by equation (13) is

$$p_{a1} = 0.074 \times \frac{2752 \times .0105}{50.78} \times \frac{549.1}{547.8}$$

$$= .0422$$

Thus, $p_1 = 0.693 + 0.042 = 0.735$ lbs. per sq. inch.

Complete conditions up to this stage are thus

$$W_a = 0.0105 \text{ lbs. per second}$$

$$W_s = 0.0612 \quad \text{,,} \quad \text{,,}$$

$$W_{s q_1} = 0.0492 \quad \text{,,} \quad \text{,,}$$

$$W_{as} = 0.0586 \quad \text{,,} \quad \text{,,}$$

$$\text{Steam} = 0.1078 \quad \text{,,} \quad \text{,,}$$

$$\text{Water} = 0.0120 \quad \text{,,} \quad \text{,,}$$

$$V_1 = 1888 \text{ ft. per second}$$

$$T_1 = 549.1^\circ \text{ F.}$$

$$t_1 = 89.5^\circ \text{ F.}$$

$$p_{s1} = 0.693 \times 144 \text{ lbs. per sq. ft.}$$

$$v_{s1} = 471.1 \text{ cu. ft. per lb.}$$

$$p_{a1} = 0.042 \times 144 \text{ lbs. per sq. ft.}$$

$$l_1 = 1039.4 \text{ B.Th.U.}$$

$$v_1 = 429.2 \text{ cu. ft. per lb.}$$

$$p_1 = p_{s1} + p_{a1} = 0.735$$

$$\text{Total volume} = 50.78 \text{ cu. ft. per second.}$$

From equation (15) we get $\gamma = 1.1353$ for the mixture, and from equation (17) $m = 1.802$. Hence, $p_2 = p_1 \times 1.802 = 1.324$ lbs. per sq. in.

$$\text{As } v_2 = v_1 \left(\frac{p_1}{p_2} \right)^{\frac{1}{\gamma}} \text{ we get } v_2 = 255.5$$

The temperature at the throat T_2 is determined by the steam pressure there. The steam pressure is $p_{s1} \times 1.802$ or 1.277 lbs. per sq. in., hence $T_2 = 569.7$ and $l_2 = 1028.3$.

Using equation (2) and assuming that the steam actually present (0.1078 lbs.) is dry saturated, the dryness fraction would become apparently 1.028, which indicates that the amount of steam is increased 2.8 per cent by the evaporation of some of the water present. Hence, the amount of steam present becomes

$$0.1078 \times 1.028 = 0.1109 \text{ lbs.}$$

and the water remaining is 0.0089 lbs.

From equation (19) the heat thus added to the steam (per lb.) is equal to the

$$\frac{\text{work done per lb. } (1 - \psi^2)}{\psi^2 J}$$

Assume that $\psi^2 = 0.85$. The work done in compressing to a pressure of 1.324 lbs. per sq. in., is 24516 ft.-lbs. per lb., hence, the added heat is

$$\frac{24516 \times 0.15}{0.85 \times 777.8} = 6.545 \text{ B.Th.U.}$$

and the amount of water evaporated is

$$\frac{6.545}{1028.3} = .006365 \text{ lbs.}$$

This increases the steam to 0.1173 lbs. and leaves

$$.0025 \text{ lbs. of water.}$$

From equation (14) we get V_2' by inserting the above value for m and find that $V_2' = 1331$ ft. per second.

The velocity is reduced by the diffuser losses and from equation (20) we get

$$V_2' = 1206 \text{ ft. per second.}$$

Thus the losses reduce the velocity about 10 per cent at the throat, and the same allowance should be made at the entry to the diffuser. The area required at the entrance to the diffuser equals

$$\frac{50.78 \times 144}{1888 \times 0.90} = 4.303 \text{ sq. ins.}$$

The specific volume of the mixture at the throat as calculated above is 255.5 cu. ft., and the throat area in sq. ins. is thus—

$$\frac{255.5 \times 0.1174 \times 144}{1206} = 3.583 \text{ sq. ins.}$$

Between the two stages, the steam used in the first stage jet would be condensed in a small jet or surface condenser in which condensate would be used as injection water in order to conserve the heat contained in the steam. Thus at the entry to the second jet we would have the same weight of air ($W_a = .0105$) as before, but it is now at a total pressure of 2.945 lbs. per sq. in. The temperature of the air and, therefore, the partial air and steam pressures are determined by the temperature of the injection water. Hence the volume of the mixture at the entry to the second stage can be calculated. The method of calculation is similar to that adopted for the first stage and will not be detailed here.

Sufficient steps of the calculation have been shown to indicate that the complete calculation is long and tedious. To show the effect of increasing quantities of steam, varying steam pressure, varying diffuser efficiencies, and other variations would require

many calculations, and naturally investigators have made more progress by actual experiment than by calculation. Given an actual ejector, the steam supplied or air extracted can be varied and the results noted.

Comparison between Isothermal and Adiabatic Compressor. It is not proposed to work out the dimensions of an isothermal compressor, but to compare it with an adiabatic compressor as regards terminal conditions and efficiency.

In the previous example, the velocities at various pressure stages in the diffuser calculated from equation (14) are as follows, the effect of the diffuser losses being omitted in this comparison—

Pressure in Diffuser. Lbs. per sq. in.	m .	Corresponding Velocity : Adiabatic.	Steam Velocity. Ft. per Second : Isothermal.
0.735	1.00	1,888	1,888
0.80	1.09	1,820	1,821
1.00	1.36	1,629	1,632
1.20	1.63	1,446	1,460
1.40	1.90	1,266	1,295

If it now be assumed that the same conditions are maintained in the entraining chamber but that an isothermal diffuser be used, the velocities as calculated by equation (22) are indicated in the last column of the above table. The exit velocity with $m = 1.90$ is thus about 2.3 per cent higher for the isothermal diffuser than for the adiabatic.

In order that the isothermal diffuser velocity should not exceed the adiabatic diffuser velocity the former should handle a greater weight of air, the extra amount being proportional to the excess of the square of the velocity. Thus, with m equal to 1.90, the extra capacity is

$$\frac{1295^2 - 1266^2}{1266^2} \text{ or } 1.047, \text{ i.e. } 4.7 \text{ per cent.}$$

This extra capacity varies with m , thus—

m .	Extra Capacity.
1.09	0.1%
1.36	0.3%
1.63	1.9%
1.90	4.7%

Efficiency of Each System. The efficiency of either method of operation considered solely as a means of compressing air is very low.

The actual work done in compressing 1 lb. of air adiabatically from a vacuum of 28.5 ins. to an absolute pressure of 1.324 lbs. by the formula

$$\frac{p_1 v_1}{\gamma - 1} \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma}{\gamma - 1}} - 1 \right]$$

is 24516 ft.-lbs. or 31.52 B.Th.U.

For isothermal compression on the work done per lb. through the same pressure range by the expression

$$p_1 v_1 \log_e \frac{p_2}{p_1}$$

is 26730 ft.-lbs., or 34.38 B.Th.U.

The efficiency expressed as a percentage for compression down to any stage ignoring entirely any heating effects is given by

$$\frac{\text{Air discharged} \times \text{work done per lb. of air} \times 100}{\text{Steam used} \times \text{heat supplied per lb. of steam}}$$

work and heat being measured in the same units.

For the adiabatic compression down to the throat the efficiency is, in this particular case, 1.52 per cent, and for the isothermal compression 1.73 per cent. Thus, whilst both efficiencies are particularly low, the isothermal efficiency is 14 per cent greater than the adiabatic.

The above efficiencies are calculated down to the throat when there was a velocity of 1,266 ft. per second remaining. If a diverging portion be added beyond the throat of the diffuser, this velocity can be used to compress the mixture further. Consider that this divergent portion is sufficient to reduce the velocity to 800 ft. per second. Then from equation (14) we get $m = \left(\frac{p}{p_1} \right) = 2.566$ or the discharge pressure is

$$0.735 \times 2.566 = 1.886 \text{ lbs per sq. in.}$$

Using this value of m in equation (22), the terminal velocity in the isothermal diffuser is 901.6 ft. per second. To compare the diffusers the terminal conditions should be the same. This higher velocity would enable the isothermal diffuser to handle $\left(\frac{901.6}{800.0} \right)^2$ times the weight of air which the adiabatic can handle. The isothermal diffuser has thus a capacity 27 per cent greater than the adiabatic, whereas in the above calculations dealing with the throat conditions the extra capacity was only 1.023^2 or about 4.7 per cent.

Dealing with the efficiencies with $m = 2.566$, we have the work done in adiabatic compression equal to 39,990 ft.-lbs., per lb., or 51.42 B.Th.U., and for isothermal compression, 42,760 ft.-lbs. or 54.98 B.Th.U. In both cases, the heat supplied is .0612 lbs. of steam, containing 356.7 B.Th.U. per lb., so that the efficiencies are

$$\text{Adiabatic } \frac{0.0105 \times 51.42 \times 100}{0.0612 \times 356.7} = 2.474 \text{ per cent.}$$

$$\text{Isothermal } \frac{0.0105 \times 1.27 \times 54.98 \times 100}{0.0612 \times 356.7} = 3.358 \text{ per cent.}$$

In this case the isothermal efficiency is about 32.6 per cent greater than the adiabatic, whereas at the throat the advantage was 14 per cent.

This calculation shows that the efficiency increases as more of the compression is considered. If the complete two stages were calculated in detail, it seems reasonable to assume that the efficiency would be increased to about 4 per cent for the adiabatic and about 5.3 per cent for the isothermal diffuser.

Any attempt to evolve a general expression, showing the relative efficiency of the adiabatic and isothermal systems of operation, would only result in a cumbersome mathematical expression. It can always be worked out as has been done above. Such a calculation only gives the relative efficiencies under one set of conditions, but is a guide for comparative purposes.

In the *Schiffbau*, (27th October, 1920), it is stated that the maximum efficiency for the adiabatic system is about 5.7 per cent.

That the efficiency is bound to be low when judged by comparison with the mechanical and electrical efficiencies of machines is evident, when it is considered that a large amount of the energy released in the jet is used to recompress the steam and discharge it against the terminal pressure, that is, the intercondenser pressure for the first stage and the atmospheric pressure for the second stage. The energy thus used is debited against the process and is not credited in any way in the calculations just made.

In all steam jet air extractors arrangements are made so that the heat contained in the steam after leaving the diffuser is transferred to the condensate. Thus, in an isothermal extractor, condensate should, if possible, be passed round the diffuser, and in both types the steam exhausted with the air would also be led into a feed heater of the surface or contact type. In this heater the steam would be condensed by condensate from the main condenser, used as cooling water.

Importance of Efficiency of Steam Jet. On this account it is usually claimed that the low efficiency of steam jet extractors, considered as air compressors, is of no serious importance from the point of view of efficient plant operation, but there is one point which should be considered. Assume that there are no losses due to leakage or radiation between the boiler, extractor, heater and boiler, and that the overall efficiency of the boiler is 80 per cent, this latter figure allowing for every loss that can be charged against the boiler, economizer, superheater (if any), feed pumps, etc. Again, assume as a concrete example that steam containing 1,000 B.Th.U. is supplied in a given time to the steam jet of an adiabatic air extractor. The efficiency being about 4.0 per cent, 40 units will be actually used in doing useful work, and 960 units returned to the boiler. In raising the steam containing 1,000 units, coal must have been used containing 1,250 units as the boiler efficiency is only 80 per cent, so that to do useful work corresponding to 40 units the net supply of heat to the boiler is (1250 - 960) or 290 units.

Every additional lb. of steam used in an air extractor, although it is afterwards condensed and its heat conserved to the system, entails a certain definite extra coal consumption. Thus it might be said that to do work corresponding to 40 units coal equivalent to 290 units has to be consumed, or the actual coal consumption is 7.25 times greater than the equivalent of the work done.

For this reason it is fallacious to claim 100 per cent efficiency for steam jet extractors as has been frequently done. Assuming radiation, leakage losses, etc., to be nil, this claim is only true as regards the steam, but not true as regards coal, which is the commodity entering into the cost sheet. Even on the steam side alone there is a degradation of heat, as the heat is necessarily returned at a lower temperature than that at which it is absorbed.

If in an isothermal compressor ejector circulating water is used to cool the diffuser, the heat rejected to the water is equal to the work done (see page 196).

Taking again a case where 1,000 units are supplied to an isothermal ejector, 53 units will be used in doing useful work and the same number rejected with the circulating water, making the balance returned with the feed water 894 units. With adiabatic compression 40 units would be used and 960 returned with the feed water. As shown earlier with the same supply of steam and similar working conditions, the isothermal compressor will handle 27.0 per cent more air than the adiabatic, but it uses 106 units against 40. Thus, if the adiabatic consumption for a given duty be 100, the isothermal

consumption including the heat losses resulting from the use of circulating water instead of condensate is

$$\frac{106 \times 100}{1.272 \times 40} = 208$$

If the boiler losses are taken into account the relative efficiency is considerably changed. To the above figures of heat used are to be added the boiler loss, assumed to be 250 heat units on the basis of a boiler efficiency of 80 per cent, making the total heat used or lost 356 units for the isothermal and 290 for the adiabatic. Again calling the adiabatic consumption 100, the isothermal consumption is

$$\frac{356 \times 100}{1.272 \times 290} = 96.5$$

Thus, under the assumed conditions the consumptions are about the same, but that depends upon boiler efficiency, etc.

When using condensate the isothermal compressor is apparently more efficient than the adiabatic. Judged from the standpoint of surface transmission of heat, the convergent diffuser is badly designed, and the cooling cannot be done efficiently. It will be necessary to use as high a mean temperature difference as possible, this requiring the coldest water available. The velocity of the water should also be high as this increases the heat transmission coefficient.

Any steam jet air extractor is most efficient when working under the conditions for which it is designed. In determining these conditions certain assumptions have to be made, particularly as to the normal weight of air to be handled. Actual figures as to the usual air allowances will be discussed later, but as the quantity may vary on either side of the allowance, the action of the ejector must be considered.

Action of the Ejector. Assume first that the quantity of air is increased, due to a leak at the turbine low-pressure gland, or to some other reason which does not increase the weight of steam passing to the main condenser to any considerable extent. The circulating water temperature will not be affected, and until the ejector removes air at the increased rate of inflow the air pressure in the condenser must increase so that an increased weight of air is contained in the condenser. This increase of pressure decreases the vacuum, which in turn slightly increases the steam consumption for equal load on the turbine. This increased pressure is also exerted in the entraining chamber in the ejector. Thus, when the air inflow is increased, the air density increases in the entraining chamber until it is so dense that

the volume removed per unit time by the jet contains a weight of air equal to the increased inflow in unit time. Thus the ejector continues working, but with a reduced vacuum. The lower velocity at the entrance to the diffuser resulting from the increased weight of air carried along with the steam causes the sectional area of the jet to increase. Successive increases in the rate of inflow of air reduce the vacuum and increase the area of the jet, and the vacuum will eventually fall so rapidly that the ejector is useless for maintaining a vacuum.

Now consider the reverse case in which the quantity of air is reduced considerably. As showing the effect most clearly, imagine that no air is to be extracted, and as two stage extractors are almost universal in use, imagine such an extractor working normally and maintaining a vacuum of, say, 28 ins., with the normal inflow of air, the vacuum in the intercooler being about 24 ins. If the air be reduced to zero the steam issuing from the first stage jet passes through the entraining chamber with no loss of velocity as no air is entrained with it. The pressure in this chamber is reduced till it corresponds to the pressure of steam at the temperature to which the condensate is cooled in the condenser. The sectional area of the jet of steam is now insufficient to fill the throat of the diffuser, and so there is no difference of pressure between the entraining chamber of the first stage and the intercooler. So that if no air gains admission, the vacuum in the entraining chamber of the second stage is also 28 ins. Exactly the same conditions would then obtain in the second stage as in the first, but as this discharges to the atmosphere, air would pass into the entraining chamber and so back into the condenser, thus spoiling the vacuum. Provided the second stage is airtight, there is no objection to the air being reduced at the first stage, and, to ensure that the second stage continues to act, many makers have fitted an automatic valve at the second stage so that air is admitted to the entraining chamber when the pressure falls below a predetermined value.

CHAPTER XIX

AIR PUMPS—HYDRAULIC TYPE CALCULATIONS

Hydraulic Air Pumps. In the class of air extractor usually designated by the term "hydraulic" a jet of water is employed as the means of extracting the air with its contained vapour and discharging it to the atmosphere. The principle of operation is very similar to that of the steam jet ejector, but there are certain variations in the method of calculating the proportions, etc.

Consider that

W_w = Weight of water flowing through the jet in lbs. per second.

h_w = Head on the water in ft.

v_w = Specific volume of water, i.e. cu. ft. per lb.

and that other symbols, as far as they are applicable, are the same as for the steam jet ejector.

The velocity of exit from the jet is

$$V_j = C\sqrt{2gh_w} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

C being a constant for the nozzle.

Equating the momenta before impact to the momentum after impact

$$\frac{W_a + W_{as}}{2g} \cdot V_c + \frac{W_w}{2g} V_j = \frac{W_a + W_{as} + W_w}{2g} V_1 \quad . \quad . \quad (2)$$

we get V_1 .

The heating effect will be found later to be of very small magnitude, so small indeed that it can be neglected without introducing any appreciable error. The water is really heated from t° , to t_1° , due to—

- (1) The loss of kinetic energy due to the mixing.
- (2) The condensation of some of the vapour contained in the air coming from the condenser.
- (3) The heat lost by the air in cooling from t_c to t_1 .

If h_c and h_1 be the heat contents per lb. of steam at the air pump suction and at the entrance to the diffuser, then

Heat lost by condensation of vapour

$$= W_{asc}h_c - W_{as1}h_1 \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The heat added, due to the conversion of kinetic energy on mixing, is

$$\frac{1}{2gJ} \left[(W_a + W_{as}) V_c^2 + W_w V_j^2 - (W_a + W_{as} + W_w) V_1^2 \right] \quad (4)$$

The heat lost by the air in cooling is

$$W_a k_p (t_c - t_1) \quad (5)$$

The sum of the last three expressions is equal to the heat gained by the water or to

$$W_w (t_1 - t_j) \quad (6)$$

If the expressions (3) to (6) be worked out for any usual case, it will be found that the increase in temperature of the water is, as stated above, very small.

As a compromise, the calculations may be made thus—

Assume that $t_1 = t_j$. Then p_{s1} and v_{s1} are known from the tables. The air temperature is the same, t_1 and the partial air pressure is obtained from

$$p_{a1} + p_{s1} = p_c \quad (7)$$

Hence v_{a1} is known from the usual equation

$$v_{a1} = \frac{53.18 \times T_1}{p_{a1} \times 144} - \frac{T_1}{2.7093 p_{a1}}$$

or if p_{a1} be measured in inches of mercury

$$v_{a1} = \frac{T_1 \times 0.7539}{p_{a1}}$$

The total air volume is $W_a v_{a1}$.

This is also the volume of the vapour, so that

$$W_{as1} = W_a \frac{v_{a1}}{v_{s1}}$$

Hence at the entrance to the diffuser we have W_w lbs. of water moving at V_1 ft. per second and $(W_a + W_{as1})$ lbs. of mixture at partial pressures p_{a1} and p_{s1} , temperature t_1 moving with the same velocity V_1 into the diffuser.

The connection between velocity of the water jet and the pressure at any point in the diffuser is given by

$$V^2 = V_j^2 - \frac{2g}{w} (p - p_1) \quad (8)$$

which is obtained from the gas equation (p. 188) in which w is a constant, so that $\int \frac{dp}{w} = \frac{p}{w}$. The area of the water jet is determined by

$$VA = W_w v_w \quad (9)$$

The compression of the saturated air follows the formula deduced earlier for the steam jet ejector. Owing to the presence of the water, the compression is done isothermally. It has been shown previously (p. 196) that in isothermal compression the heat rejected by the compressed gas to the cooling medium is equal to the work done in compression. The whole of the energy required for the actual work of compression, and the heat rejected, is obtained from the kinetic energy contained in the moving fluid at the entrance to the diffuser. The work done per lb. of air in compressing the air from p_1 to p_3 lbs. per sq. ft. is given by the equation given on page 206. This, then, is also the heat rejected to the cooler. Calling this w_a we have the kinetic energy lost equal to $2w_a$. Hence, if V_3 be velocity at exit, we get

$$\frac{W_a + W_{as} + W_s}{2g} (V_1^2 - V_3^2) = 2w_a \quad . \quad . \quad (10)$$

By adding energy w_a to the jet water as heat, the temperature is raised to t'_3 so that

$$W_w (t'_3 - t_1) = w_a \quad . \quad . \quad . \quad . \quad . \quad (11)$$

This ignores the heat required to raise the temperature of the air in contact with the water. This heat is obviously small in comparison with the heat taken up by the water, as the mass of the water present is so much greater than the air, and the specific heat of air is only 0.2375 against unity for water. For all practical purposes it will be sufficiently accurate to determine t'_3 by equation (11), and then from the steam tables obtain p_{s3} , etc. The volume of the air is obtainable from the usual equation knowing p_{a3} and t'_3 . Hence the weight of vapour in the air at exit is

$$\frac{W_a v_{a3}}{v_{s3}} = W_{as3} \quad . \quad . \quad . \quad . \quad . \quad (12)$$

A second and closer approximation to t_3 can be made by correcting for the heat of condensation given up by the vapour to the water. This heat raises the water temperature to t_3 as

$$W_w (t_3 - t'_3) = W_{as1} h_1 - W_{as3} h_3 \quad . \quad . \quad . \quad (13)$$

Leblanc Rotary Pump. In this type of pump, of which a more detailed description will be given later, a jet of water impinges on to the inner circumference of a rotating ring of blades as shown diagrammatically in Fig. 94. By the action of these blades, the water from the jet is divided into sheets which are propelled by their kinetic energy down a pipe leading to a tank open to the atmosphere.

As β and V_o are known, and as W_1 is known from the relation

$$W_1^2 = V_1^2 + v^2$$

as the angle BOA is a right angle it is possible to calculate W in terms of V_o . Without working out any example it is obvious that W is greater than V_o , hence in Fig. 96 AC is greater than the arc AB.

In Fig. 96 the lines inclined at an angle of β to the tangents at the limits of the arc subtended by the jet are produced to meet at O, and the angle COB is bisected by OM. Then OM is the mean direction of motion of the layer or sheet of water. At the inter-

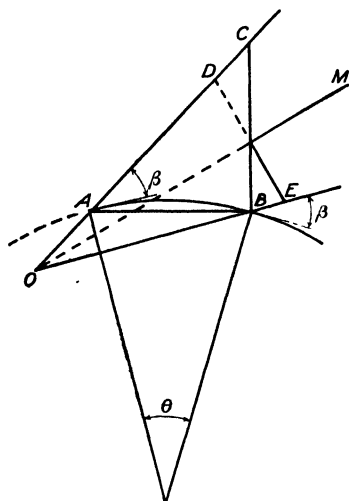


FIG. 96.

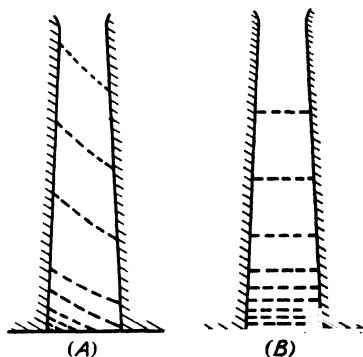


FIG. 97.

section OM with BC draw a line at right angles to OM. The water which first left the blade is the distance CD in front of the mean line, and the last water to leave is BE behind the mean line.

Hence, OM being the axis of the diffuser, the sheets of water would be inclined to the axis of the diffuser as shown in Fig. 97. It would be better to have the sheets at right angles to the axis of the tube as shown at B, as the area of each sheet being smaller, the thickness is greater with B than with A. The actual displacement is the same in both cases.

In order to obtain the form indicated in Fig. 97 (B), M. Maurice Leblanc has shown* the necessary relations between the speed of the water and the rotating blades as follows. In Fig. 95 let $r_o =$

* See *La Lumière Électrique*, July 16 and 23, 1910.

NB, the outer radius of the row of blades, and V_o , W , θ , and β the same meaning as before.

In Fig. 95 we have $NA = NB = r$

$$\angle ANB = \theta$$

AD is the tangent at A and $\angle CAD = \beta$

OB is at the same angle β to the tangent at B

$$\text{hence } \angle AOB = \theta.$$

We therefore get that $\angle CAB = \beta + \frac{\theta}{2}$ and also

$$\angle CBO = \frac{\pi}{2} - \frac{\theta}{2} \text{ and } \angle ABO = \beta - \frac{\theta}{2}$$

$$\text{Hence } \angle CBA = \angle CBO - \angle ABO$$

$$= \frac{\pi}{2} - \frac{\theta}{2} - \beta + \frac{\theta}{2}$$

$$= \frac{\pi}{2} - \beta$$

Considering the triangle ABC we have

$$AC \cos \left(\beta + \frac{\theta}{2} \right) + BC \cos \left(\frac{\pi}{2} - \beta \right) = AB \quad . \quad (15)$$

and also

$$\frac{AC}{\sin \left(\frac{\pi}{2} - \beta \right)} = \frac{BC}{\sin \left(\beta + \frac{\theta}{2} \right)}$$

$$\text{whence } BC = AC \frac{\sin \left(\beta + \frac{\theta}{2} \right)}{\cos \beta}$$

Substituting in equation (15) and simplifying we get

$$AC = AB \frac{\cos \beta}{\cos \frac{\theta}{2}}$$

From triangle ABN we get

$$AB = 2NA \sin \frac{\theta}{2} = 2r_o \sin \frac{\theta}{2}$$

$$\text{Hence } AC = 2r_o \tan \frac{\theta}{2} \cos \beta$$

The water which left the wheel at the point A has travelled to C when the last portion of that sheet is leaving the wheel at the point B, so we get by equating the times taken

$$\frac{AC}{W} = \frac{r_o \theta}{V_o}$$

$$\text{or } \frac{2r_o \tan \frac{\theta}{2} \cos \beta}{W} = \frac{r_o \theta}{V_o}$$

As θ is always comparatively very small it is sufficiently accurate to assume that

$$\frac{\theta}{2} = \tan \frac{\theta}{2}$$

and from this we get

$$W = V_o \cos \beta.$$

Substituting this value for W in equation (14) we get

$$V_o^2 (\cos^2 \beta - 1) - W_1^2 - 2 V_o W_1 \cos \beta = 0 \quad (16)$$

The initial data are h_w , V_1 and V_o . From h_w we get v , W_1 is obtained from V_1 and v_1 by the equation given at the top of page 215, hence the only unknown in equation is β . The required value can be obtained by trial and error.

As an example, assume that the inner and outer radii of the blade wheel are 9 ins. and 12 ins. respectively, and that the speed of rotation is 1,500 r.p.m. This gives values of 117.9 and 157.2 ft. per second for V_1 and V_o . Assume that the effective head on the jet is 6 ft. plus 28 in. vacuum, so that the velocity v is 49.28 ft. per second. From equation, page 215, $W_1 = 127.7$. Substituting known values in equation (16) and simplifying we get

$$\cos^2 \beta - 1.625 \cos \beta - 1.6598 = 0$$

Obviously $\cos \beta$ must be negative and β is therefore in excess of 90° . Trying $\cos \beta = -0.7$ and -0.8 , it will be seen that the required value is $\cos \beta = -0.7103$. Hence β is approximately 135° .

In this particular case the outer edges of the blades are to be bent back 45° beyond the radial as indicated in Fig. 98.

For the same case α is such that

$$\sin \alpha = \frac{v}{V_i} = \frac{49.28}{117.9} = 0.418, \text{ hence } \alpha = 24.7^\circ$$

The effect of a drop in vacuum upon the working of the pump should be noted. The water flows through the stationary nozzle due to the positive head on the supply side, and also to the suction

head corresponding to the vacuum maintained in the condenser and the pump. Thus, with 6 ft. head on the nozzle and a 28 in. vacuum, the effective head is $\left(6 + \frac{28 \times 13.6}{12}\right)$ or 37.73 ft. If a sudden rush

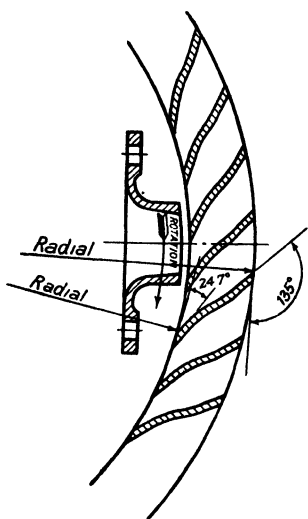


FIG. 98

of air lowers the vacuum by, say, 4 ins., the head is reduced to 33.2 ft., and the quantity of water passing is reduced

in the ratio of one to $\sqrt{\frac{33.2}{37.73}}$ or 0.938;

a reduction of 6.2 per cent. A further loss takes place owing to the fact that the angles of the blades on the rotating impeller are fixed.

Reproducing in Fig. 99 the first rectangle of velocities in Fig. 94, suppose that v is reduced from the original value to OA' . OB is unchanged and so the resultant velocity, assuming that there was no constraint, would be OC' , the angle BOC' being less than angle BOC or α . But as α is fixed by the blades the velocity of the water relative to the blade must be along OC . Resolving OC' along and at right angles to OC , the velocity of the water relative to the blades is OD , and $C'D$ is the velocity at right angles to the blade, which is wasted in shock at entry, causing eddies and heating.

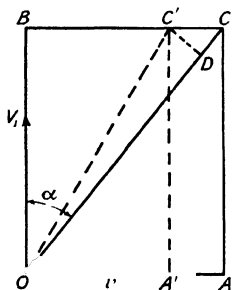


FIG. 99.

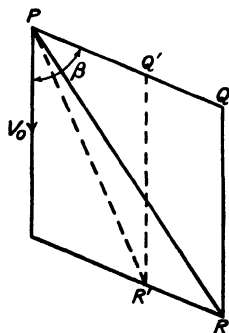


FIG. 100.

Referring back to Fig. 94, and reproducing the second velocity rectangle in Fig. 100, the new value for W_1 is $PQ' = OP$ and the

resultant is PR' . This velocity W is reduced and its direction of motion is changed. The previous value of β was used to determine the angle of the diffuser in relation to the axis of the stationary nozzle, so that with the reduced value the diffuser is no longer in the most advantageous position.

Thus, a drop in vacuum below the conditions for which the pump is designed, which may be caused by a sudden rush of air, militates against efficient operation in three ways—

- (1) The quantity of water passing is reduced.
- (2) The velocity of efflux is reduced, and
- (3) The direction of efflux is no longer co-axial with the diffuser.

With less water passing and the sheets of water inclined at a greater angle to the axis of the diffuser, thus increasing the area of each sheet, the operation is less stable as the sheets will not retain their plane formation so readily.

These reasons caused this type of pump to be regarded as only suitable for dealing with normal air quantities. A method of overcoming this difficulty is described on page 273.

CHAPTER XX

AIR PUMPS—DESCRIPTION OF RECIPROCATING TYPES

The Edwards Air Pump. As the earliest type of air pump to be used, the reciprocating pump is naturally the first to be described. But it is not used extensively now for large plants, and its field of action is rather restricted to low vacuum plant. The Edwards type of air pump is still largely used, and a representative pump is illustrated in Fig. 101, which shows a three-cylinder pump, $14\frac{1}{2}$ ins. diameter by 12 ins. stroke, running at 110 r.p.m. Edwards pumps are usually of the two- or three-throw (or cylinder) type, except for marine installations, where a single-barrel pump was practically universal with the reciprocating steam engine fitted in the usual tramp steamer. In Fig. 101 the air pump suction is shown at *A*, both air and condensate entering the pump by the same branch. This opening leads into the space marked *B*, surrounding the air pump barrels, and the air enters the barrel through the ports *C*, when these ports are uncovered by the pistons or buckets on their down strokes. The condensate entering by *A* flows down through the passages *D* to the space under the buckets *E* and collects in the conical bottom. The underside of the bucket is shaped similarly to the space *E*, and the clearance at the bottom of the stroke is comparatively small. As soon as the point of the bucket enters the water which has collected in the bottom of space *E* the water is scattered and, guided by the shape of the chamber *E*, passes through the ports *C* on to the upper side of the bucket. The ports *C* are closed by the up-stroke before the water can return through the ports, and this is carried up with the bucket, being discharged with the air through the head valves *F*.

These valves are usually of the Kinghorn or other metallic type, although thick rubber is also used. The Kinghorn valve is shown in Fig. 101. The valve is of the direct-lift type, and consists of two thin steel plates which are forced up by the escaping air against a light spring. The area through the grids under the valves is usually about 30 per cent of the area of the buckets. Rubber valves, when used, were gripped in the centre as shown in Fig. 102, a guard being provided to limit the lift in both types. This figure also shows an alternative design of Kinghorn valve without spring as in Fig. 101.

The air escapes to atmosphere through pipe *G*, Fig. 101, a funnel

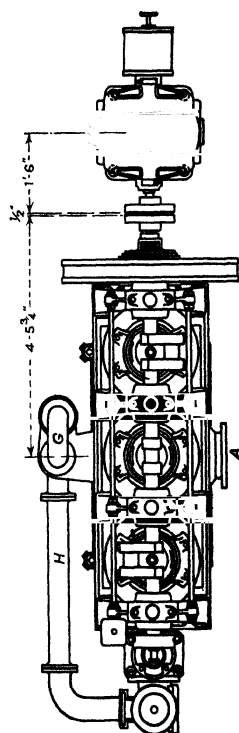
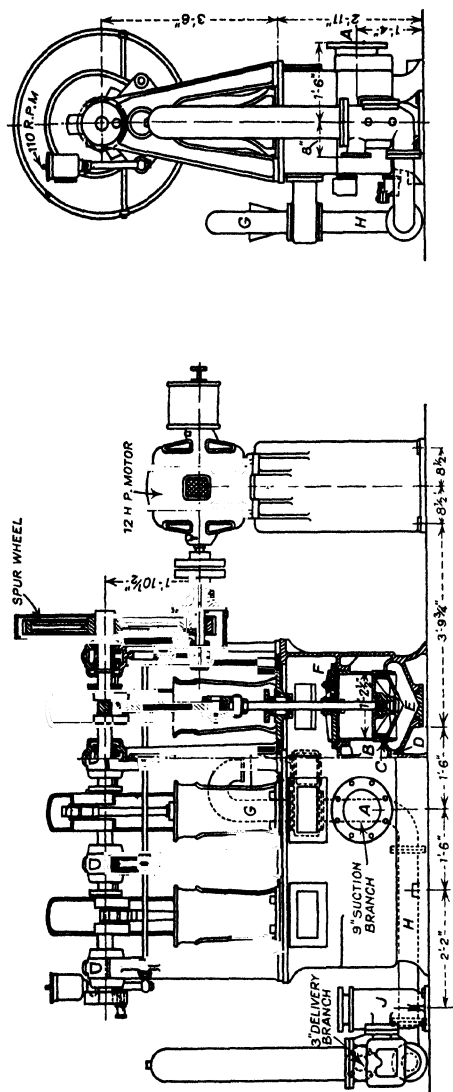


FIG. 101.—ARRANGEMENT OF 14½" × 12" AIR PUMP.
(Cole, Marchant & Morley, Ltd.)

being provided under the outlet to allow the moisture carried over by the air to drain back into the pump. The condensate is taken away by pipe *H* to a force pump *J*, driven from an overhung crank. This pump discharges the condensate to the hot well.

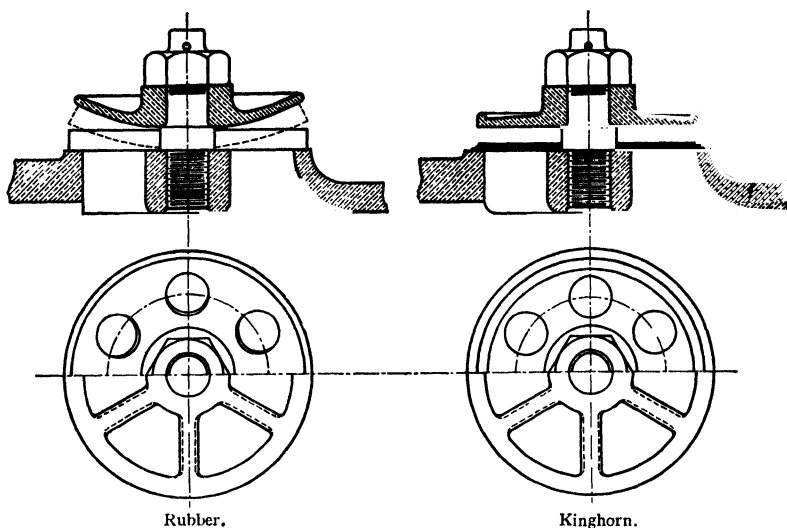


FIG. 102.
VALVES OF EDWARDS PUMP.

Fig. 101 shows a pump driven by an electric motor through gearing, the method usually adopted. Some engineers prefer a direct-coupled motor, but as the pump speed rarely exceeds 140 r.p.m., this is expensive. A cheaper arrangement was to mount the pinion directly on the motor shaft, and to omit the outer bearing for the motor shaft shown under the end-bearing of the pump shaft in Fig. 101.

The construction of the pump calls for no further comment. The piston speed of this type should not exceed 200 to 250 ft. per minute, the speed of the pump shown in Fig. 101 being 220 ft. per minute. Low piston speeds are necessary to obtain high efficiencies, due to the necessity of there being an appreciable pause at the end of each stroke to enable the air to flow into the barrels.

With the maximum piston speed fixed, the displacement required per minute determined the dimensions of the barrels, but the displacement required per lb. of steam was apparently determined by rule of thumb based upon past experience. The Edwards Air

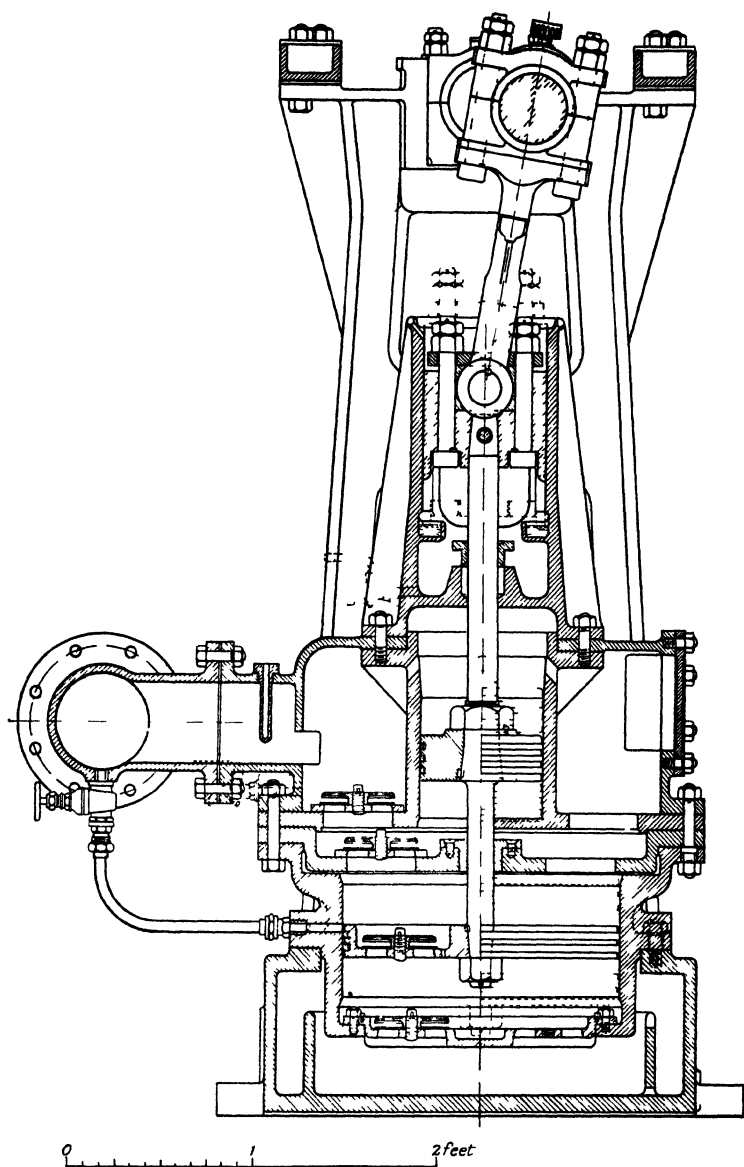


FIG 103
COMPOUND AIR PUMP
(C A Parsons & Co Ltd)

Pump Syndicate used to recommend the following displacements per lb. of steam for the various vacua—

26 in.	.	.	.	0.65	cubic feet per lb. of steam
27 in.	.	.	.	0.90	" " " "
28 in.	.	.	.	1.10	" " " "

It is now customary to design for vacua of 29.0 ins. and 29.1 ins., but a vacuum of 28 ins. was considered to be the economical limit for those pumps, and the above schedule terminated at that figure.

In the *North-East Coast Institution of Engineers and Shipbuilders Report* to which reference has been made, the rate at which condensate was passed through the receiver *A* in Fig. 87 was varied, but as the condensate had been deaerated an increase in the quantity of condensate did not increase correspondingly the quantity of air which entered mainly by the nozzle *G*. In test No. 2, of which some details have been given, the condensate rate was 12,200 lbs. per hour. Tests Nos. 10 and 18 only differed from No. 2 in the condensate rate being increased to 42,000 and 58,700 lbs. per hour respectively. The vacua with a 2 mm. air nozzle in each case were 28.30 ins., 27.95 ins. and 27.75 ins. The displacements per lb. of condensate were 1.052, 0.309 and 0.224 cu. ft. respectively. The latter figures are very much lower than the figures given above as the recommendations of the Edwards Air Pump Syndicate. At that time practically no attention was given to the air contents of the steam reaching the condenser, and the relative proportions of air and condensate in test No. 2 probably represented the proportions usually met then. In this test the 2 mm. nozzle would pass 5.856 lbs. of air per hour, according to the formula used in the report, this corresponding to 4.89 lbs. of air per 10,000 lbs. of steam per hour. In test No. 18 the air supply was only 1.00 lbs. per 10,000 lbs. of steam, hence the vacuum maintained was very much in excess of that corresponding to a displacement of 0.224 cu. ft. per lb. on the basis of the above recommendations. In connection with the weight of air per 10,000 lbs. of steam see remarks on pp. 355-8.

The Parsons Compound Air Pump. The Edwards type of pump was used very extensively, but in Fig. 103 there is shown a reciprocating pump which was used very largely by Messrs. C. A. Parsons & Co., Ltd., in connection with their condensing plants for turbines.

The Parsons air pump was of the compound type, both cylinders being on the same centre line. The lower cylinder is the first stage, and is provided with foot and head valves of the light direct lift type, similar valves being provided on the piston. The upper and smaller cylinder forms the second stage. The head valves of the first stage act as foot valves to the second stage.

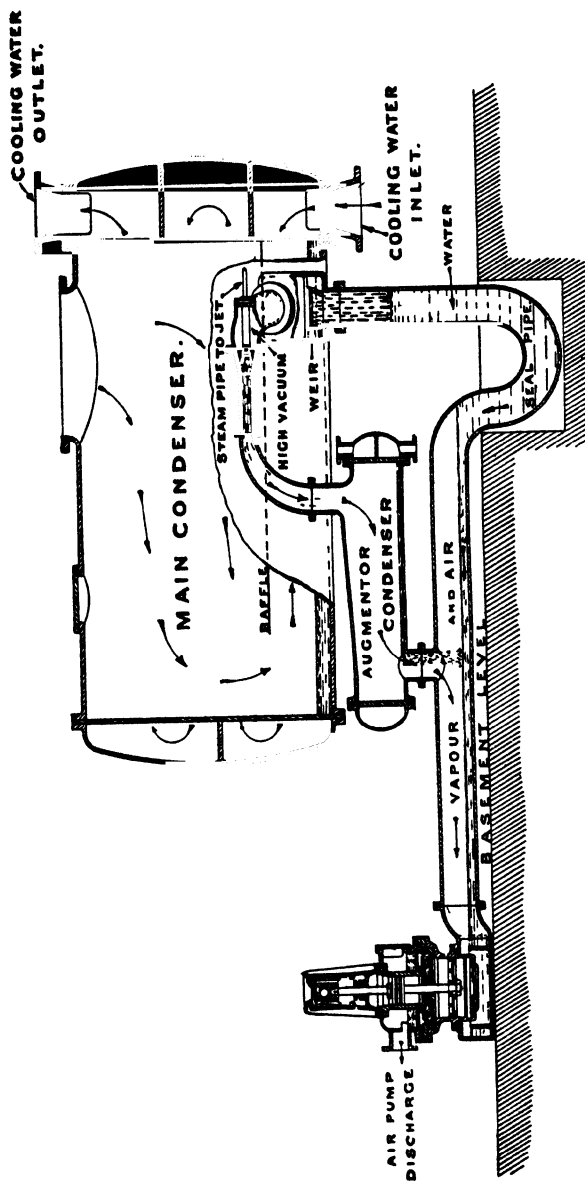


FIG 104
 VACUUM AUGMENTOR ARRANGEMENT
 (C A Parsons & Co., Ltd.)

Parsons Vacuum Augmentor. Parsons vacuum augmentor is illustrated diagrammatically in Fig 104. In this the separate extraction of the air and condensate from the condenser is shown, the air being withdrawn by the steam jet and the condensate going to the air pump via the seal pipe. Both air and condensate are combined again at the higher pressure created by the jet and dealt with in a Parsons compound air pump as shown. The vacuum

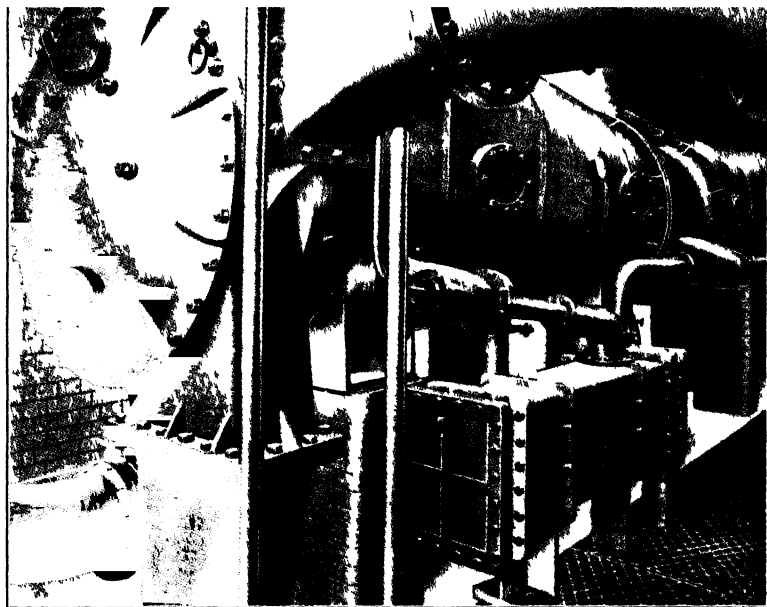


FIG 105

VACUUM AUGMENTOR PLANT

(C A L I E S & C O 111)

augmentor system was also used extensively with Edwards type pumps. In Fig 105 there is shown a typical old installation of a small augmentor plant. Undoubtedly condensing plants fitted with vacuum augmentors were an advance upon the then current practice, but as the steam turbine has ousted the reciprocating engine, so the rotary pump eventually replaced the reciprocating air pumps in the majority of installations. When other means of obtaining high vacua in large units were established, the Parsons vacuum augmentor was criticized on account of all the heat in the steam used in the jet being rejected to the circulating water, and so being lost to the system. Notwithstanding this defect, the Parsons

vacuum augmentor played, in its time, an important part in the development of air extractors, and permitted reciprocating pumps to be used for larger generating units than was previously practicable.

Wet and Dry Air Pumps. As stated earlier, one of the first changes made to secure greater air pump efficiency was to work the pumps on the so-called wet and dry system. The pump would be similar to that illustrated in Fig. 101, but the connections and divisions were as shown diagrammatically in Fig. 106.

In this arrangement barrels A and B deal with the air only, but water has to be circulated through these barrels to seal them, and

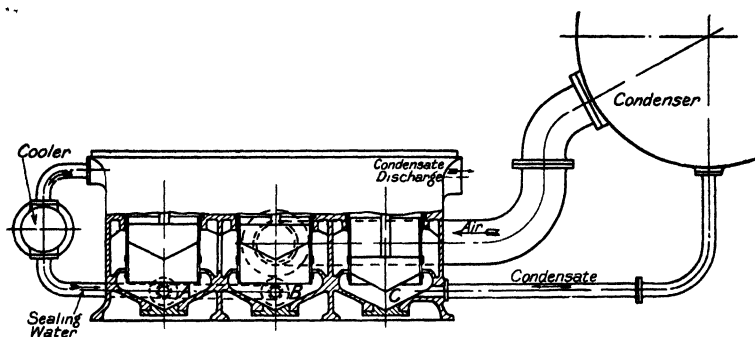


FIG. 106
WET AND DRY AIR PUMPS

this is passed through a small cooler as shown. This cooler is worked as a shunt on the circulating water system. The remaining barrel, C, draws the condensate from the bottom of the condenser and discharges it to the hot well. The space above these head valves is divided from the similar space above barrels A and B by a partition.

With this arrangement the displacement per lb. of air or steam, assuming that the same size pumps are used as in the older arrangement, is only two-thirds as great as in the older method, but it was usually found that the vacuum was quite as good, with the advantage that the condensate was not unduly cooled, as was the case with the usual wet pump arrangement. This was due to the sealing water which approximated in temperature to the circulating water inlet cooling the air at the dry pump suction. This has the two-fold effect of reducing the air volume and of raising the relative capacity of the pump by reducing the partial vapour pressure. The temperature of the condensate handled by the wet barrel C

approximates to the discharge temperature of the circulating water, the difference being dependent upon the drainage in the condenser.

Dry Air Pumps. A reciprocating air pump of the dry type was largely used at one time before the present day demand for steam jet air ejectors. In this pump the air cylinder is really dry, and not dry by name only as in the so-called "wet and dry" arrangement, described on page 227. Such pumps are really air compressors working up to the atmospheric pressure, whereas compressors as usually understood work from the atmospheric pressure upwards.

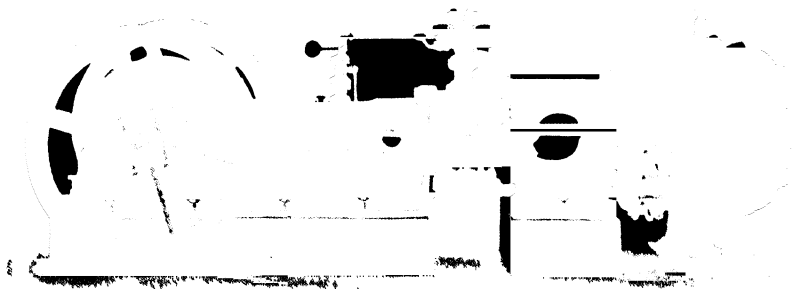


FIG. 107

DRY AIR PUMP

(Worthington Simpson, Ltd)

As an example of this type of pump, Fig. 107 illustrates a horizontal steam-driven dry vacuum pump as made by the Worthington Pump & Machinery Corporation and Worthington-Simpson, Ltd. A section through the air cylinder is shown in Fig. 108. This is of the two-stage single cylinder type. Air from the condenser enters by the valve A and is discharged on the return stroke at the same pressure through valve B. From here it passes to the other end of the cylinder and is drawn in through the valve C. It is then finally discharged to atmosphere through valve D.

For this two-stage single cylinder pump it is claimed that the volumetric efficiency is high, due to low clearances, and further that there are two sets of valves between the condenser and the atmosphere.

In any reciprocating pump, the volumetric displacement depends upon the rotative speed of the pump which, in turn, is limited by the type of valve used. In this particular pump the Worthington Feather valve is used in which the moving part is reduced to the absolute minimum.

THE WORTHINGTON FEATHER VALVE. This valve consists of three parts, the seat, the flexible steel valves, and the valve guard.

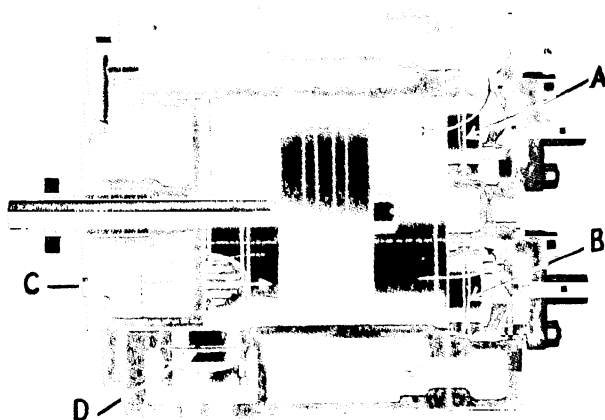


FIG. 108.
SECTION THROUGH PUMP CYLINDER.
(Worthington-Simpson, Ltd.)

The three parts are shown separately in Fig. 109. On the left there is the seat with ground faces. In the centre, are shown the thin steel strips which form the valves. This steel is extremely flexible.

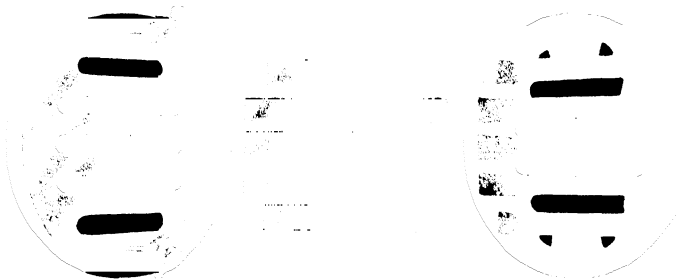


FIG. 109
STEEL FEATHER VALVE.

On the right is the valve guard, having iron bars corresponding with the openings in the valve seat, but concave to the seat. The action of the valve is indicated more clearly in Fig. 110, showing a section. In this, A is the seat and B the valve guard. The spring C is shown in the raised or discharge position, and it will be noticed that the ends are in contact with the seat. When the discharge

stroke ends the spring returns to the flat position, the ends going further into the slots provided in the guard.

With this valve, due to the small moving weights and the absence of valve operating mechanism, the rotative speed can be high, and speeds up to 560 r.p.m. have been used in compressors.

This valve is reversible and can be used either for suction or discharge, as the valve guard fits into the seating provided for the valve seat.

Dry air pumps of this type are very economical in power absorbed. A two-stage motor driven pump with Worthington Feather valves

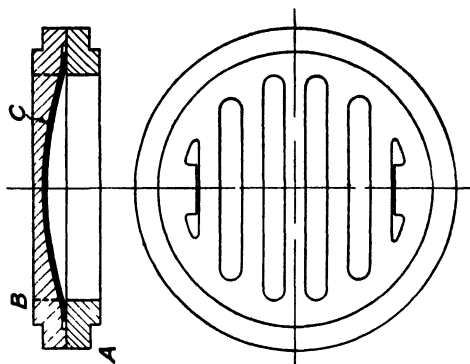


FIG. 110
STEEL FEATHER VALVE
(Worthington Simpson, Ltd.)

has been installed at Cedar Rapids, Iowa, for the Iowa Railway & Light Co., with a 18,000 sq. ft. condenser.* With a steam load of 95,752 lbs., a vacuum of 29.54 ins. was maintained, and the input to the motor was only 12 k.w. With a load of 129,713 lbs. and a vacuum of 29.4 ins., the input was 13 k.w. The output of the main turbine was probably not less than 10,000 k.w., so that the air pump power was not greater than 0.13 per cent.

This type of air pump was also made in the single stage single cylinder type, a section through a cylinder being shown in Fig. 108.

Dry reciprocating air pumps were usually driven by steam engines, the air cylinder being tandem to the steam cylinder as shown in Fig. 107, but other methods of driving could be adopted, such as belt or motors, etc.

* See *Report, Prime Movers Committee*, National Electric Light Association, 1922.

Pump Valve of The Mirrlees-Watson Co. Another type of valve adopted for use with a dry air pump is shown in Fig. 112. This valve has been used largely by The Mirrlees-Watson Co., Ltd.

It will be seen that it is a slide valve fitted with two spring-controlled discharge valves mounted on the back of the slide valve. The action of the valve is clearly explained by reference to the

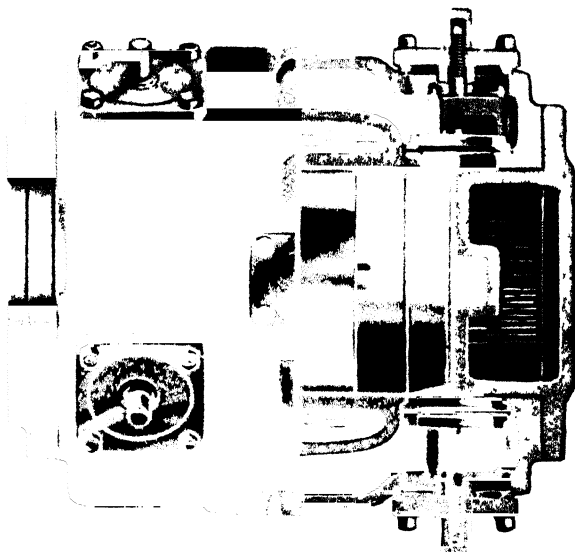
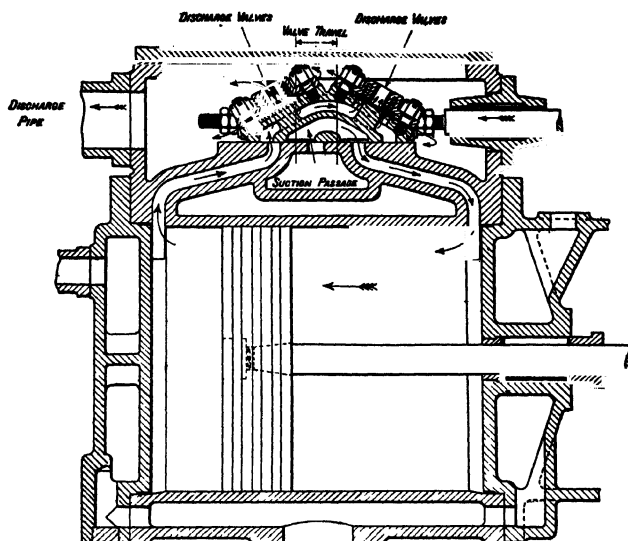


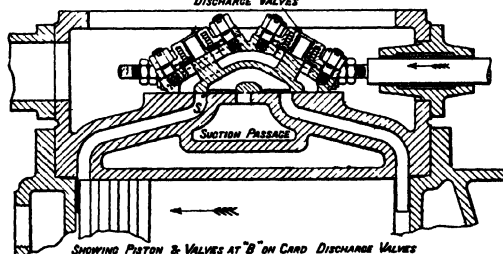
FIG. 111

indicator diagram also shown in Fig. 112. The upper view in Fig. 109 shows the position of piston and valve when the point A on the diagram has been reached. Air is being discharged through both discharge valves which have just opened. In the middle view the piston is nearly at the end of the stroke, and the discharge valves have closed as the piston is nearly stationary. The right-hand end of the cylinder is closed to the suction passage, and is just on the point of being connected to the left-hand end through the equalizing port marked S. In the bottom view the piston has just commenced the return stroke, and the pressure on both sides of the piston is equal to the pressure in the evacuated chamber. A little later the two ends of the cylinder are no longer in connection and the normal suction and compression strokes recommence.

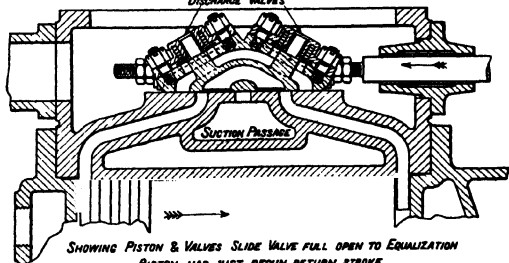
As will be seen from the indicator diagram, this valve enables the cylinder to commence drawing on the evacuated chamber at once,



SHOWING POSITION OF PISTON & VALVES AT "A" ON CARD
DISCHARGE VALVES HAVING JUST OPENED
DISCHARGE VALVES



SHOWING PISTON & VALVES AT "B" ON CARD DISCHARGE VALVES
HAVE JUST CLOSED SLIDE VALVE IS JUST OPENING TO EQUALIZATION AT S
DISCHARGE VALVES



SHOWING PISTON & VALVES SLIDE VALVE FULL OPEN TO EQUALIZATION
PISTON HAS JUST BEGUN RETURN STROKE

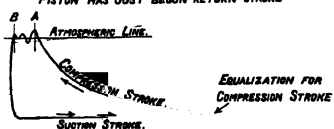


FIG. 112.—DRY SLIDE VALVE AIR PUMP.
(Mirrlees-Watson & Co., Ltd.)

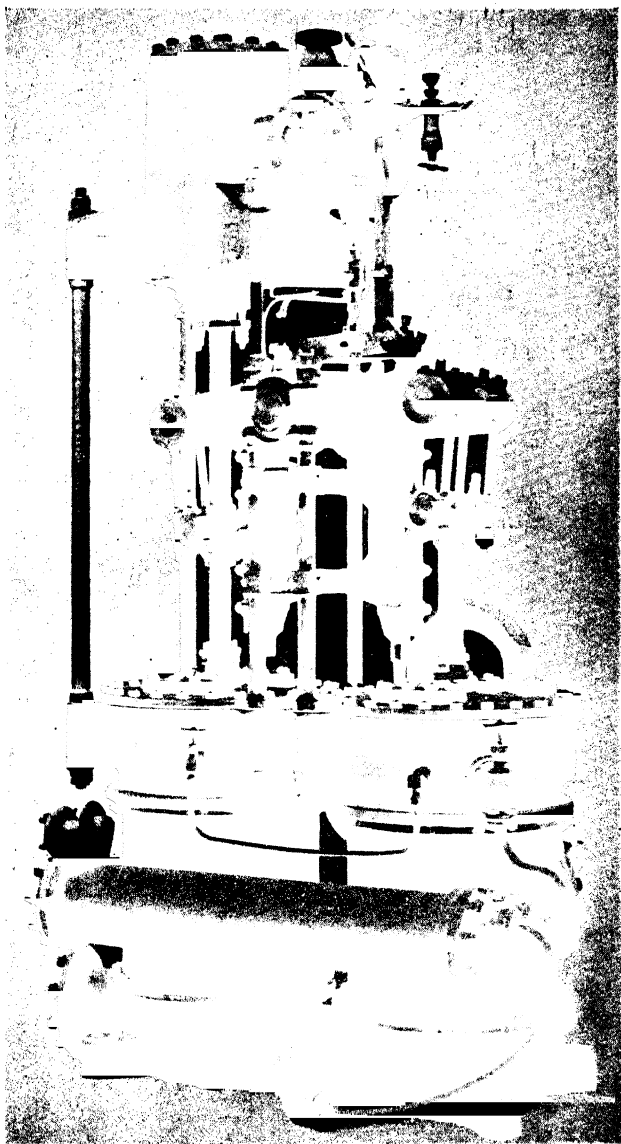


FIG. 113.
DUAL AIR PUMP.
(G. & J. Weir, Ltd.)

and no fraction of the stroke is wasted in expanding the air remaining in the clearance space down to the suction pressure.

The Weir "Dual" Air Pump. A reciprocating air pump of the wet and dry type as made by Messrs. G. & J. Weir is illustrated in Fig. 113.

This pump has been named by the makers the "Dual" air pump, as it is both wet and dry. It is operated by a single steam cylinder having the wet pump on the same line as the steam cylinder as this

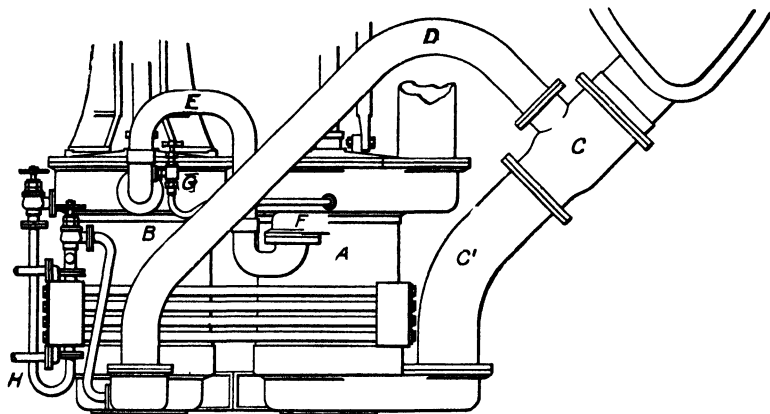


FIG. 114

pump has the greater load. The dry pump is driven by a beam and links as shown.

As shown in Fig. 114, there is only one connection from the condenser C, but a branch on this pipe allows the air to be drawn away to the dry pump on the left, the condensate passing down through C' to the wet pump. Both pumps are of the three-valve type, having foot, bucket, and head valves. The dry pump discharge is led by the pipe E to the barrel of the wet pump just under the head valves, where it enters through a spring-loaded valve F. This valve F is so loaded that it opens when the pressure in the dry pump exceeds 10 lbs. absolute pressure, so that the dry pump works between the condenser vacuum of, say, 28 ins. and 20 ins. This small pressure range increases the capacity of the dry air pump as compared to what it would be if discharging against the atmospheric pressure. To ensure the operation of the dry pump, sealing water must be supplied, and for this purpose condensate is aspirated when starting up by opening valve G. Condensate then passes to the head valve space in the dry pump, and from there it passes through the cooler

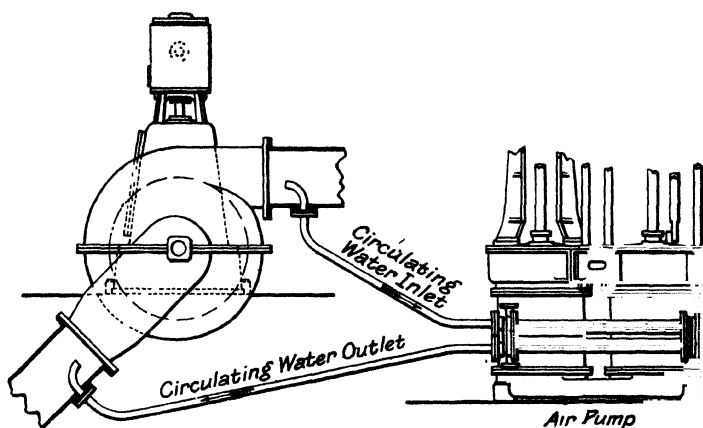


FIG. 115
 COOLER CONNECTIONS FOR "DUAL" AIR PUMP
 (G & J. Weir, Ltd.)



FIG. 116.
 MARINE TYPE OF WET AND DRY AIR PUMP.
 (Worthington-Simpson, Ltd.)

and to the dry air pump suction. The difference in pressure of 8 ins. between the top and bottom of the dry air pump is sufficient to force the sealing water through the cooler.

Returning to Fig. 113, the cooler will be seen in the front with circulating water connections at the left-hand end, and the sealing water at the right-hand end, with pipe leading the sealing water to the dry pump suction. Above the cooler will be seen a small pipe for admitting condensate to the dry pump on starting up the plant.

The cooler is a shunt on the main circulating water system, the recommended type of connection being shown in Fig. 115.

For this pump it is claimed that a higher vacuum can be obtained than with the usual all wet pump, for the reasons given when dealing with the Edwards wet and dry pump arrangement.

Another pump working on a similar cycle is shown in Fig. 116, this being made by Worthington-Simpson, Ltd., for an important passenger liner. This figure shows the cooler, and the connection from the dry air cylinder to the wet cylinder into which the dry air passes between the piston and the head valves.

CHAPTER XXI

AIR PUMPS—DESCRIPTION OF STEAM EJECTOR TYPES

Steam Ejector Air Pumps—Development. Steam ejector air pumps are very extensively used at the present time, and a large amount of ingenuity has been exercised in introducing new and improved forms. Inasmuch as all depend upon the kinetic energy of a jet of steam, there cannot be any basic difference in the various types but merely in the application of this common principle.

The Parsons vacuum augmentor, already described, appears to represent the first practical application of the use of a steam jet for withdrawing air from a condenser. In the Kinetic system, due to the late Mr. D. B. Morison, the second stage reciprocating pump used by Sir Charles Parsons was replaced by an hydraulic air pump. In all forms of pumps that are usually classified as steam ejector air pumps, the second stage pump as well as the first stage pump consists of a steam ejector, so that the whole air extracting system is free from mechanically moving parts. Steam jets had been used for moving air earlier than these applications, an example being the use of jets for forced draught in marine and other boilers.

The Breguet "Ejector." One of the earliest air extractors to rely upon steam jets alone was that introduced by Maison Breguet, Paris, and manufactured in the United Kingdom by Messrs. Hick, Hargreaves & Co., Ltd. A section through this type of ejector (termed by the makers an "ejectair") is shown in Fig. 117. The arrangement is practically self-explanatory, consisting of the first stage jet B and the second stage jet E, with an inter-condenser of the jet type C. Water supplied to the inter-condenser is sprayed through the jet D, the nozzle being of simple design and arranged to give a whirling action to the spray, thus ensuring maximum condensing and cooling effect.

The auxiliary condenser C is drained by means of a loop-pipe which, in addition to providing an effective drain for the auxiliary condenser, at the same time maintains the required difference of absolute pressure between the main and auxiliary condenser.

At the side of the second stage jet, the patent stabilizing valve H is fitted. This is provided in order to ensure stability under very light loads. As explained earlier (pages 197 and 198), with a very small quantity of air passing through the air extractor, the throat

of the combining tube of the second stage would not be maintained sufficiently full to prevent air leaking back into the auxiliary condenser C.

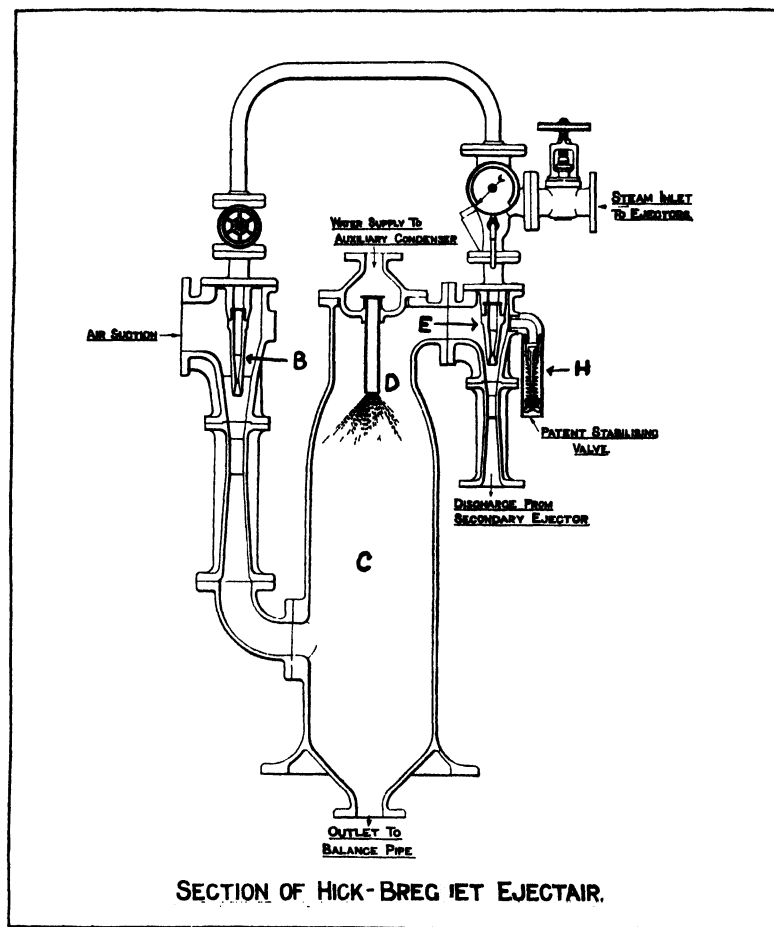


FIG. 117

The stabilizing valve is so arranged that when the vacuum in the auxiliary condenser becomes so high that the ratio of compression by the second stage is near the critical, the stabilizing valve opens slightly, admitting just the amount of air required to maintain stability.

From these remarks it will be gathered that the stabilizing valve

on many plants only comes into action at very rare intervals, so that the whole of the energy of the steam supplied to the jets is utilized in removing the air and other gases from the main condenser. The regulation of this valve is of the simplest possible, and enables the makers to claim absolute stability under any condition of working for their apparatus.

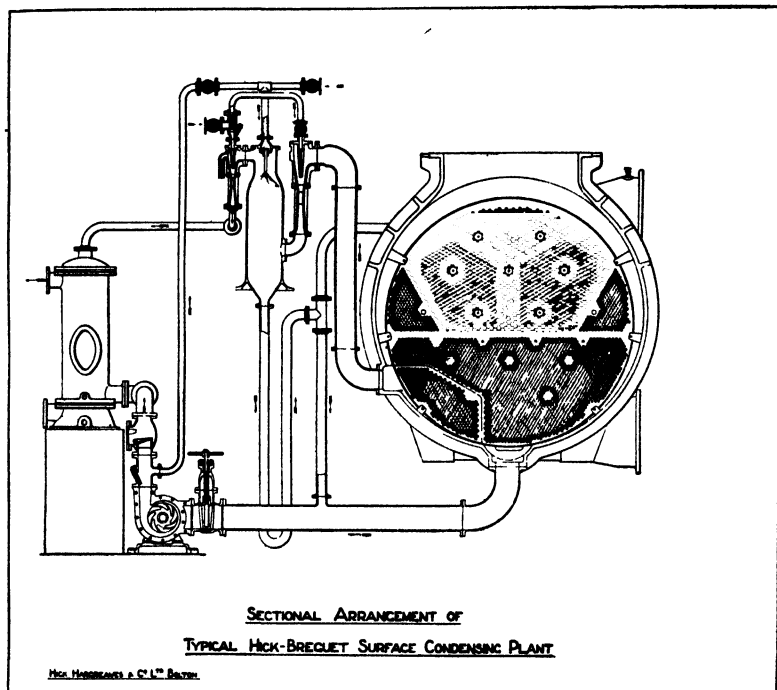


FIG. 118

The steam consumption of the ejector is not effected in any way by the fitting of this stabilizing valve, this being entirely a question of the area of steam nozzles for a given duty. A sectional view of a condenser fitted with this type of air ejector is shown in Fig. 118. This figure also shows the usual arrangement of feed heater for taking the steam from the second stage jet and also the usual arrangement of extraction pump connected thereto, the arrangement ensuring high overall thermal efficiency of the plant, the heat in the steam from the second stage jet being added to the condensate on its way to the feed tanks. The loop pipe draining the auxiliary condenser is also clearly shown.

It will be noted that an alternative supply of water is shown to the auxiliary condenser placed between the steam jets. This is for the use of town's water or other source, pending a sufficient supply of condensate being available from the main condenser. This auxiliary supply is shut off immediately there is a fair load on the machine.

The Mirrlees-Leblanc Ejector. An early form of steam jet ejector is shown in Fig. 119. The steam is divided at each stage

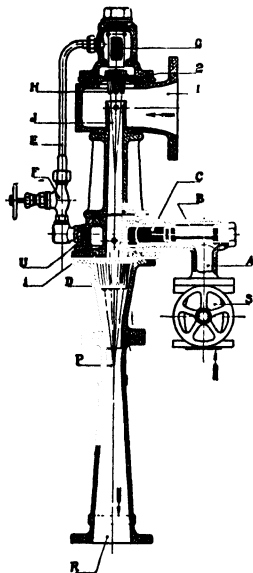


FIG. 119.

MIRRLEES-LEBLANC
SIMPLE EJECTOR.

into a number of jets all directed into the tail pipe. The controlling idea is to cause the steam to present as great a surface as possible, so that air could be more easily carried out by surface friction. Thus if, instead of a single jet with any specified area, an alternative design used, say, n smaller jets with the same total area all directed towards the throat of the tail pipe, the aggregate perimeter in the multi-jet apparatus would be $\sqrt{n}$ times the perimeter in the single jet apparatus. Thus the surface which draws the air out by friction is $\sqrt{n}$ times as great, due to the division into n jets. A further advantage is claimed in that whilst air between a moving jet and a stationary wall must have a lower velocity adjacent to the wall, or in other words must experience a certain slip, in a multi-jet apparatus the slip will be reduced, as a great proportion of the air is between two or more moving jets and is thus dragged forward on all sides and not retarded by the wall.

The variation in practice between those designers who use single and multi-jets with competitive apparatus, indicates that this question is not of paramount importance. If this be so, then the single large jet is preferable as it is not so liable to be choked, being larger than any one of the multi-jets and, further, it is cheaper to construct. The latter point is of importance, as the price obtained for any plant is determined by the cheapest efficient construction, hence the multi-jet must be sold at the same price as the single jet with a slightly smaller margin of profit.

One advantage claimed for this design at that time was the absence of any inter-condenser, as without it, it was possible to maintain

rectilinear motion throughout from the air inlet to the steam and air discharge. Undoubtedly, this was of some advantage, but the saving in steam consumption resulting from the use of an inter-condenser was of greater value, as has been proved by the almost universal present use of an inter-condenser.

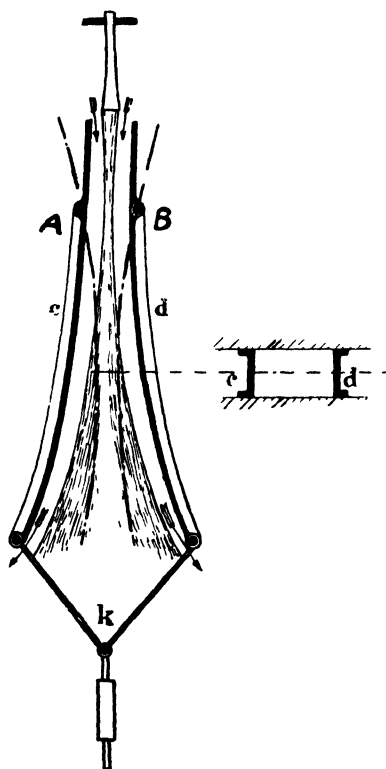


FIG. 120

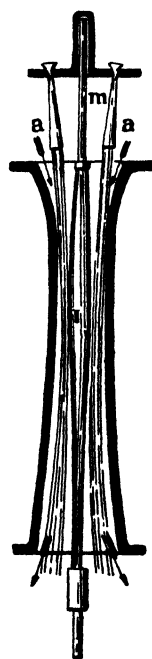


FIG. 121

LEBLANC EXPERIMENTAL DESIGNS

Experimental Designs to Facilitate Starting. A special feature of the Leblanc extractor is the atmospheric connection shown at the collar just above the throat of the second stage tail pipe. On page 198 it has been shown that when starting up the area required in the tail pipe is much greater than when the vacuum is established. M. Maurice Leblanc tried to provide a tail pipe in which the area could be varied when in operation. One device is shown in Fig. 120. In this the tail pipe was of rectangular section with two opposite

walls pivoted at points A and B. For starting, the upper ends of the wall were moved towards each other, but when the vacuum was established they were moved outwards, causing the area to be reduced considerably. Another arrangement is shown in Fig 121, in which a double tapered rod is capable of axial adjustment through the tail pipe. For starting, the smallest diameter of the rod is in the way of the throat, but the largest diameter is in that

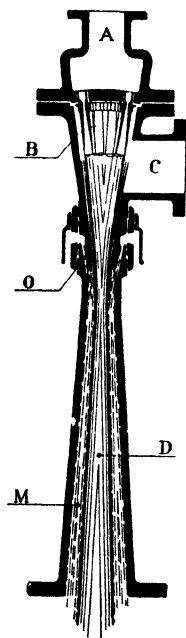


FIG 122—LEBLANC
EXPERIMENTAL
DESIGN

situation when the normal vacuum is obtained. Leblanc found that these methods were only suitable for laboratory apparatus, but he also found that a similar effect could be obtained by making a connection to the atmosphere at a point just in front of the throat of the second stage tail pipe. The argument justifying this method is as follows. When starting up, the jet of steam together with the entrained air occupies the whole area at any section of the tail pipe, but as the motion of the jet becomes more rapid owing to the decreased rate of air extraction and greater initial velocity of the steam resulting from the reduced pressure into which it is discharged, the moving jet becomes smaller in section and tends to leave the walls of the tail pipe. The arrangement is shown more clearly in Fig 122. When starting, the pressure at O and at the discharge of the tail pipe is the same, or practically the same, but as soon as a partial vacuum is established air tends to flow into the tail pipe at O. Until the jet leaves the wall at O air cannot enter, but the moment this takes place air does enter and is carried out by the moving jet of steam. Thus air packing is used to fill the space left by the jet owing to its increasing velocity. It would appear that as the velocity increases, the throat of the jet becomes smaller and moves lower down the tail pipe, packing-air entering in such quantities as are required to fill the tail pipe. If the jet leaves the tail pipe wall above the throat, air will pass back into the entraining chamber. As to whether this arrangement presents any advantage over that adopted in the Breguet ejector is probably a matter regarding which the respective partisans could expound, but the practical effect appears to be the same.

Metropolitan-Vickers Leblanc Type Ejector. Another form of the

Leblanc ejector of the simple type as made by the Metropolitan-Vickers Electrical Co., Ltd., is seen in Fig. 123. This is substantially similar to that shown in Fig. 119.

When used with an inter-cooler, the ejector as made by the latter

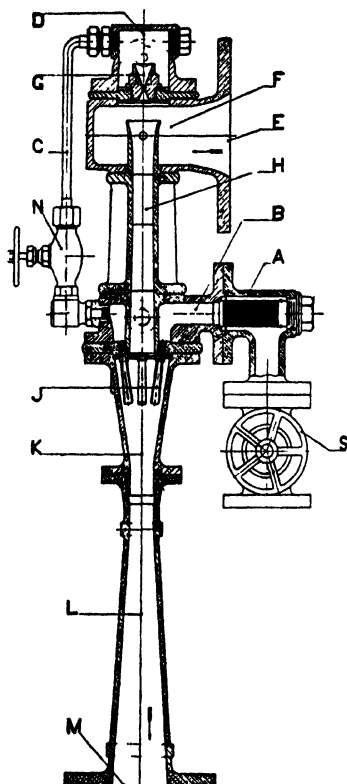


FIG 123

METROPOLITAN-VICKERS
LEBLANC EJECTOR

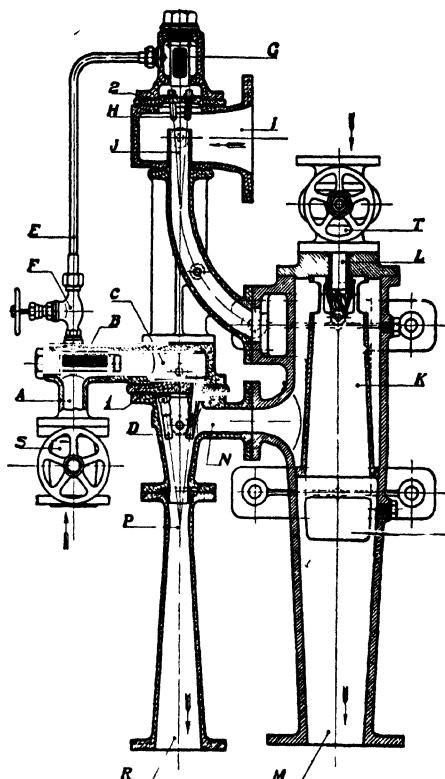


FIG 124

METROPOLITAN-VICKERS LEBLANC
TYPE EJECTOR

firm assumes the form shown in Fig. 124. The two stages are maintained in the same relative positions as in the simple type of ejector, but the discharge from the first stage is diverted to a small jet condenser at one side. The mixed air and steam from the first stage enter the condenser by the port V. The cooling water enters the condenser at the top and is given a whirling motion by the scroll shown. The cooling water leaves at the bottom by flange M, which is connected to a loop pipe leading to the condenser, and the air

rises through the falling cooling water. The air passes into the outer annular chamber and so down to the entry to the second stage.

The Metropolitan-Vickers Electrical Co., Ltd., have developed a design of steam jet air ejectors fitted with surface inter-condensers, and after-condensers as shown in Fig. 125. From this figure, the first and second stage steam nozzles B_1 and B_2 with their combining chambers, tail pipes, etc., can be seen. Condensate is used to condense the jet steam and enters by the flange L_1 . The cooling

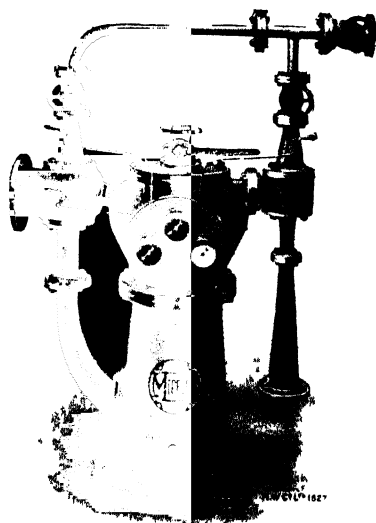


FIG. 127

MIRRLEES-WATSON TWO-STAGE EJECTOR

tubes are of the Field type, and condensate passes up the inner tubes and down through the annular space between the concentric tubes to the water box P_1 , whence it passes through the tubes in the second or after condenser to the outlet L_2 .

Mirrlees-Watson "Central Condenser" Type. Another form of two-stage ejector with inter-condenser is shown in Fig. 126, this being made by Messrs. The Mirrlees-Watson Co., Ltd., and termed by them a "Central Condenser" type. In this, it will be noticed that a single jet has been used in each stage and a comparatively large jet condenser between the stages. The tail pipe of the first stage does not increase greatly above the throat diameter, indicating that the throat velocity is not much in excess of the critical velocity. The

large inter-condenser should ensure complete condensation and enable the air to become thoroughly separated from the water.

An external view of this ejector is shown in Fig. 127. The three gauges mounted on the ejector indicate the steam pressure at each stage and the vacuum maintained at the condenser connection.

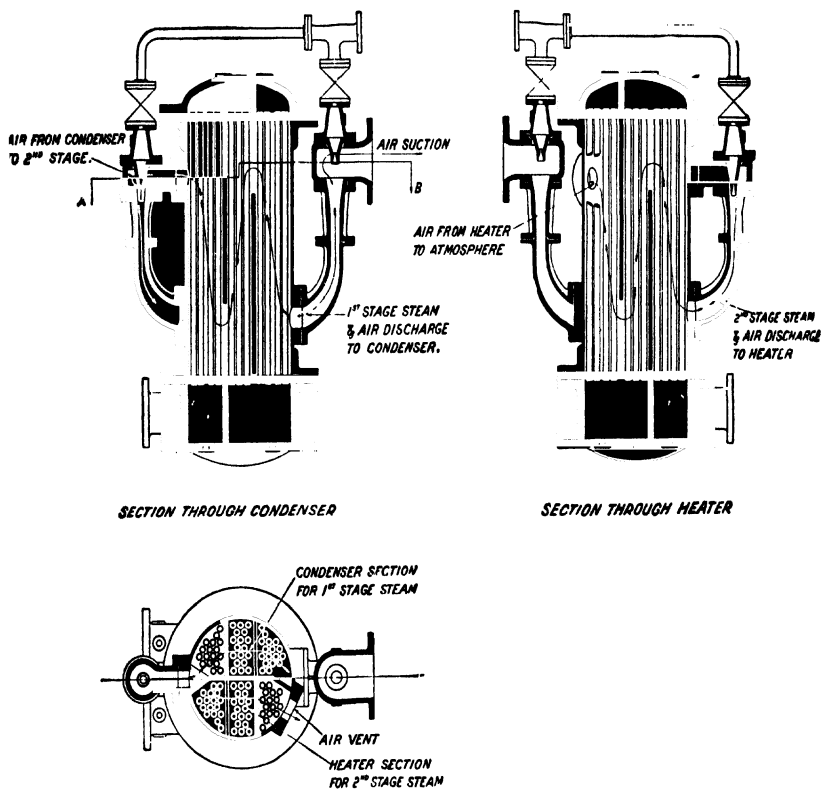


FIG. 128

MIRRLEES-WATSON EJECTOR WITH INTER-CONDENSER

A similar two-stage ejector fitted with a surface inter-condenser and a surface after-heater also made by The Mirrlees-Watson Co., Ltd., is shown in Fig. 128.

The Wheeler Multiple Jet Ejector. In Fig. 129 there is illustrated a form of steam jet air pump, placed upon the American market by the Wheeler Condenser & Engineering Co., Ltd. This is of the two-stage type with inter-condenser of the jet type. In

this ejector the multiple form of jet is used. Condensate is introduced through the opening H and sprays into the central chamber in the upper part of the jet condenser. The heated condensate

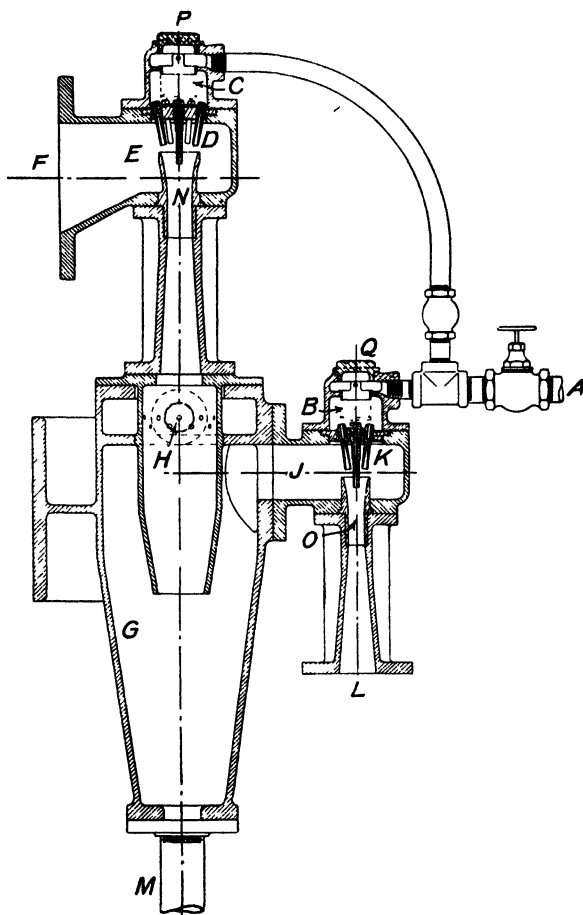


FIG. 129

STEAM JET EJECTOR

(Wheeler Condenser & Engineering Co., Ltd.)

passes away from the bottom of the inter-condenser to the main condenser hot well, and the air rises to the entrance to the second stage jet. Fig. 130 shows an external view of one of these ejectors. The bracket shown is provided so that the ejector can be bolted directly to the condenser, thus occupying no floor space.

In any such arrangement it is essential to arrange that the ejector is sufficiently high so that the necessary length of loop may be obtained to ensure proper operation. In Fig. 131 the loop pipe is shown diagrammatically. A is the connection to the inter-condenser, and B the condensate discharge to the condenser or extraction

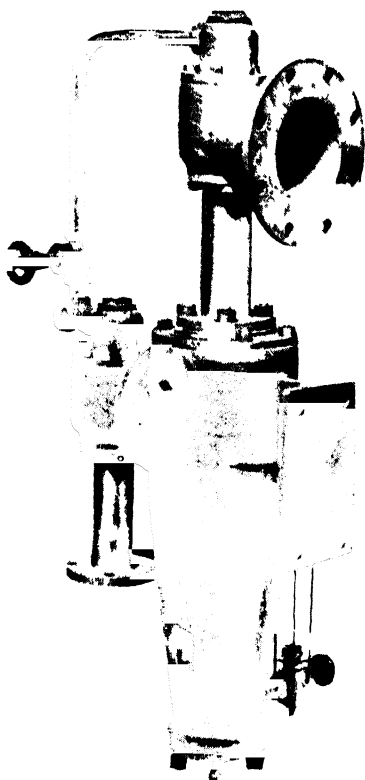


FIG 130

STEAM JET EJECTOR.

(Wheeler Condenser & Engineering Co., Ltd.)

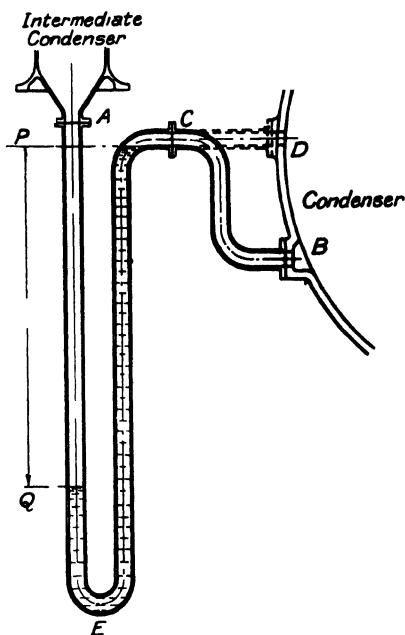


FIG 131.

LOOP TYPE CONNECTION TO
CONDENSER

pump suction. The highest point C is connected to the condenser or to the air suction pipe leading to the ejector. In some arrangements the third leg is omitted and the condensate discharge placed at D. The pressure at C being that in the condenser, the water level in the first leg must be such that the difference PQ is equal to the pressure or "plenum" created in the first stage. Thus, the height between A and E must correspond to a pressure in excess of the maximum pressure ever built up by the first stage.

As the vacuum in the inter-condenser is usually not lower than 24 ins. Hg. the maximum value of PQ in Fig. 128 will not exceed about 6 ft., so that the full length of the loop to retain a suitable margin should be about 8 ft.

With a surface inter-condenser, the problem of returning the main

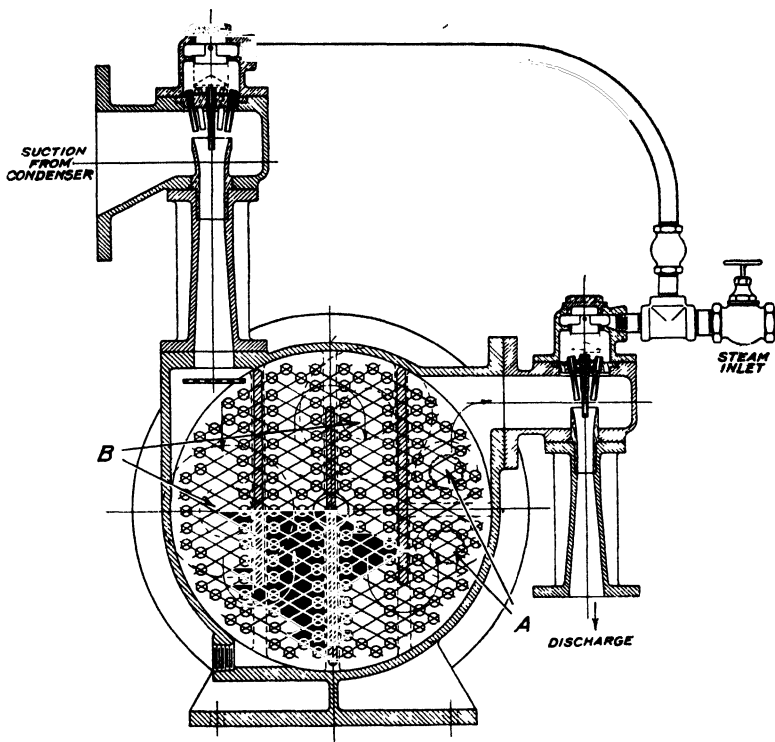


FIG 132

STEAM EJECTOR WITH SURFACE INTER-CONDENSER.

(Wheeler Condenser & Engineering Co., Ltd.)

condensate to the system does not arise, as the inter-condenser can be on the discharge side of the condensate extraction pump.

There is, however, the water of condensation formed in the inter-condenser which it is desirable to save, and this can be done through a loop pipe as described above.

When fitted with surface inter-condenser the Wheeler ejector becomes as shown in Fig. 132. The proportioning of the surface inter-condenser has to be on much more generous lines than practice

with main condensers would indicate, on account of the high ratio of air to steam in the inter-condenser when compared with the average ratio in main condensers. As has been indicated when discussing condensers, the tubes nearer the air suction are less effective as heat transmitters owing to the high thermal resistance of the air, so, in the inter-condenser, it may be said that it is entirely composed of surface comparable to or even worse than the least efficient portion of the main condenser. Ample surface has to be provided and, a matter of great importance, the velocity of the combined steam and air must be kept as high as possible. In the Wheeler inter-condenser it will be noted that this is done by providing suitable baffles giving the steam and air the maximum length of passage from entrance to exit.

As, when starting up there is no condensate to pass through the inter-condenser, a portion of the tubes, those nearest to the second stage suction, are served with circulating water as a shunt on the main circulating water system. When in operation with normal load, this supply of circulating water can be closed down as it removes some heat which might otherwise be conserved. Should the load on the steam turbine be subject to severe fluctuations it may be advisable to leave the circulating water supply in use to ensure that the inter-condenser continues to operate. The circulating water inlet and discharge flanges are marked A, and the main condensate inlet and discharge flanges B in Fig. 132.

HEAT ECONOMY IN SECOND STAGE. In order to conserve the heat used in the second stage jet it is usual to discharge the air and steam into a special sealing tank, or to condense the steam in a small condenser. If a jet condenser or sealing tank be used with condensate from the main condenser as injection or sealing water, the condensate becomes saturated with air which is a great disadvantage, owing to the fact that the oxygen in the contained air causes pitting of the boiler and economizer tubes. With the higher steam pressure now generally used, economizer tubes are sometimes made of steel instead of the usual cast iron, and this emphasizes the necessity of eliminating air from the system. As in order to conserve the heat in the second stage steam, main condensate must be used to condense it, the only satisfactory solution is to use a surface after-condenser.

COMBINED AFTER-CONDENSER. The Wheeler Condenser Co. have combined the inter-condenser and the after-condenser in one casing, as shown in Fig. 133. In the latter the after-condenser is shown under the inter-condenser. Entirely separate, of course, on the steam side, the two condensers, or compartments are in series on

the water (or main condensate) side, the condensate passing first through the inter-condenser. The steam temperature in the inter-condenser is much lower than in the after-condenser, so that the coldest condensate should pass through the inter-condenser. Although, in the majority of two-stage ejectors, more steam is used in the second stage than in the first, it will be noted that the surface of the after-condenser is less than that of the inter-condenser. This is accounted for by the higher steam temperature in the after-condenser, giving a greater mean temperature difference with a correspondingly greater heat transmission per sq. ft. per hour.

The condensate passes away from the after-condenser by the bottom connection shown in Fig. 133, and the air and any other non-condensed gases escapes to the atmosphere by an orifice not indicated. The condensate formed in this after-condenser can be led to any open tank, such as that belonging to a water measurer of the open-weir type. If no such tank is available, and an open tank is not possible if a closed feed system is adopted (see Chapter XXVI), special means have to be adopted to get this condensate back into

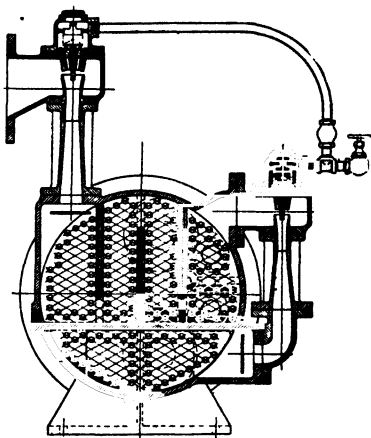


FIG. 133.

COMBINED INTER- AND AFTER-
CONDENSER.

(Wheeler Condenser & Engineering Co., Ltd.)

the system. To use a loop pipe as in the case of the first stage is impracticable, as the loop would have to be about 32 to 34 ft. long, which would only be possible with barometric plant.

VACUUM TRAP. To overcome this trouble the Wheeler Condenser Co. have introduced a vacuum trap which is illustrated in section in Fig. 134. Traps operating on a similar principle have been adopted by other makers. The essential part of this is a float-operated valve, G. As the float rises, due to condensate from the after-condenser B passing through C into the chamber D, a port is uncovered in G. This port is V-shaped and is developed so that the opening is a function of the rotation of the valve G, thus depending on the height of the float. This arrangement ensures that the port opening is drowned, whatever may be the rate of inflow of condensate. For every rate of steady inflow of

condensate there is a corresponding height of the float. Thus, as the opening is always drowned so that air cannot pass, the condensate, after passing through the valve G, can be led via K to any part of the main condensate system even under vacuum. It will be noted that an overflow connection L is provided should a tube burst in the after-condenser.

The Wheeler "Radojet" Ejector. A steam jet ejector which has been used extensively in the United States and is now being made in Great Britain by John Musgrave & Sons, Ltd., is that introduced by the C. H. Wheeler Manufacturing Co. This is illustrated in

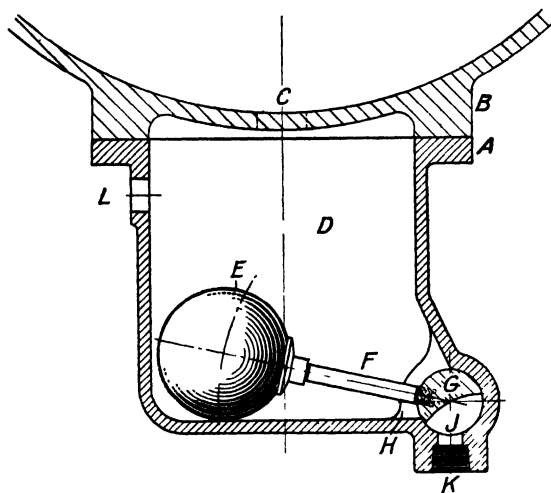


FIG. 134.

VACUUM TRAP.

(Wheeler Condenser & Engineering Co., Ltd.)

section in Fig. 135. From this it will be seen that it is of the usual two-stage type, with multi-jets in the first stage. The second stage is peculiar in that the jet flows radially in the form of a sheet or disc from the central nozzle into the concentric diffuser.

This radial flow of the steam after passing the nozzle point has given rise to the name "Radojet," applied to this type of apparatus by the makers.

The action of the first stage is obvious from the illustration. The discharge from the first stage is led by a bifurcated pipe to each side of the disc of rapidly moving steam. The steam is admitted from the connection B at the left-hand side of the illustration, and expands through the nozzle C of which the throat D is designed to

pass the required amount of steam. The expansion part of the nozzle subsequent to the throat consists of a short length between the throat and the end of the nozzle tube and the radial space between the end of the nozzle tube and the nozzle disc. This point is adjustable, but once the best position is determined for the actual working conditions, no further adjustment is needed and the end of the nozzle disc spindle is covered by the dome F. Adjustment of the nozzle disc axially with the nozzle does not affect the throat so that the amount of steam passing is not altered. This adjustment does change the sectional area of the exit end of the nozzle, this being equal to the perimeter of the nozzle disc by the distance between the disc and the nozzle end. Thus, adjustment of the nozzle disc is equivalent to altering the angle of the divergent part of a convergent divergent nozzle of the usual type, but does not alter either the throat area or the length.

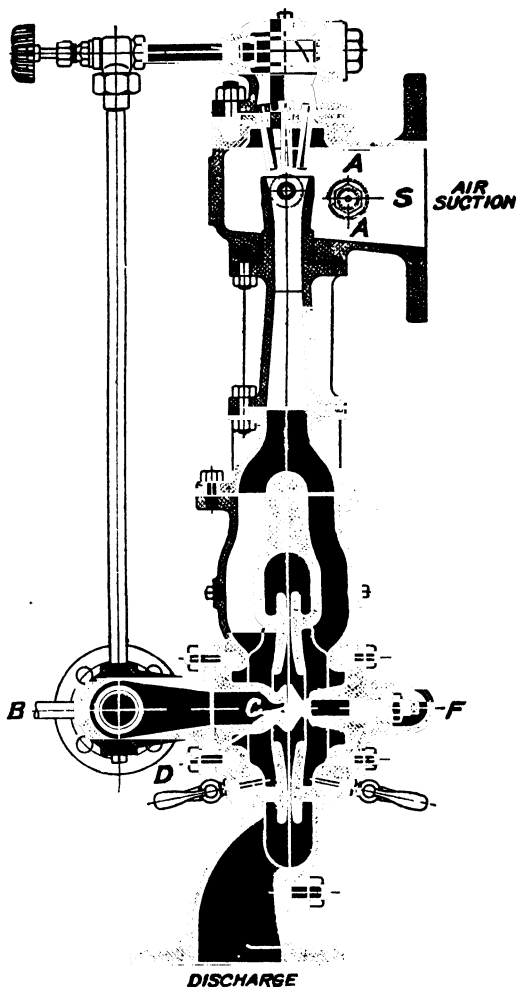


FIG 135.

WHEELER RADOJET' EJECTOR.

Judged as a steam nozzle, this arrangement is comparatively inefficient when it is considered that steam moving at a velocity which may reach 4,000 ft. per second has to turn through a right

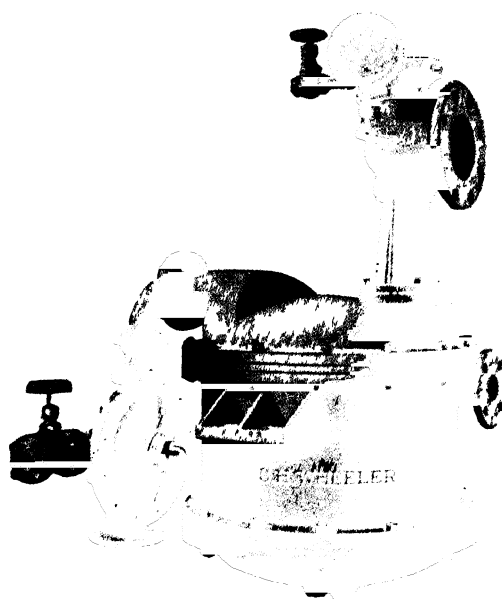


FIG. 137

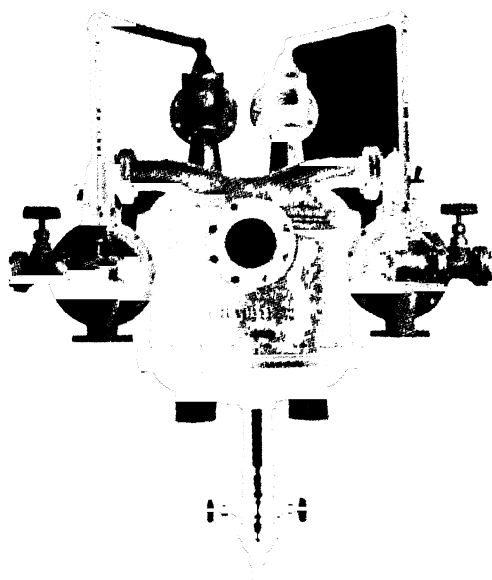


FIG. 138

TWIN RADOJET EJECTORS WITH COMBINLED INTER-CONDENSER.
(C. H. Wheeler Manufacturing Co.)

air, be carried into chamber E and fall on to the tube plate. On account of the high ferrules shown, this condensate cannot re-enter tubes *t*, and naturally passes into the drain tubes *d*. These are provided with a water seal at the lower end through which the condensate passes into chamber C, and hence through pipe *p*, to the condensate trap or loop pipe. Due to the velocity of the air through the tubes A in the second pass, there is a difference of pressure between chambers C and E against which the water seal has to operate. Without the water seal, tubes *d* would act merely as condensing tubes *t*. In the third pass, from chambers E to F,

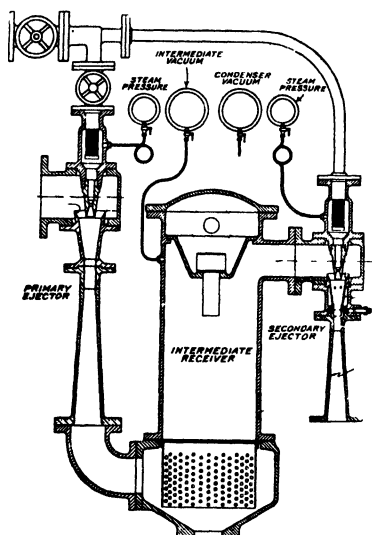


FIG. 139.

KINETIC STEAM EJECTOR
(Contraflo Condenser Co., Ltd.)

both air and condensate are flowing in the same direction, hence the absence of water seal, tube *d* taking some of the condensate collecting in chamber E. Pass four is similar in action to pass two. The second stage ejector is similar to that used in the simple two-stage ejector shown in Fig. 135.

The external appearance of these ejectors is shown in Fig. 137. In larger units, it is usual to provide two or more ejectors and, if of the inter-condenser type, two can be mounted on one inter-condenser as shown in Fig. 138.

The Kinetic Air Pump. The Kinetic system, handled by the

Contraflo Condenser Co., Ltd., was originally a combination of steam and water ejectors working in series, but they have developed purely steam ejectors which will be dealt with here. An illustration of the arrangement is shown in Fig. 139. In principle it will be seen to be similar to other arrangements previously described. There are the usual two stages with an inter-condenser of the jet type, the water from the inter-condenser being led by a loop pipe

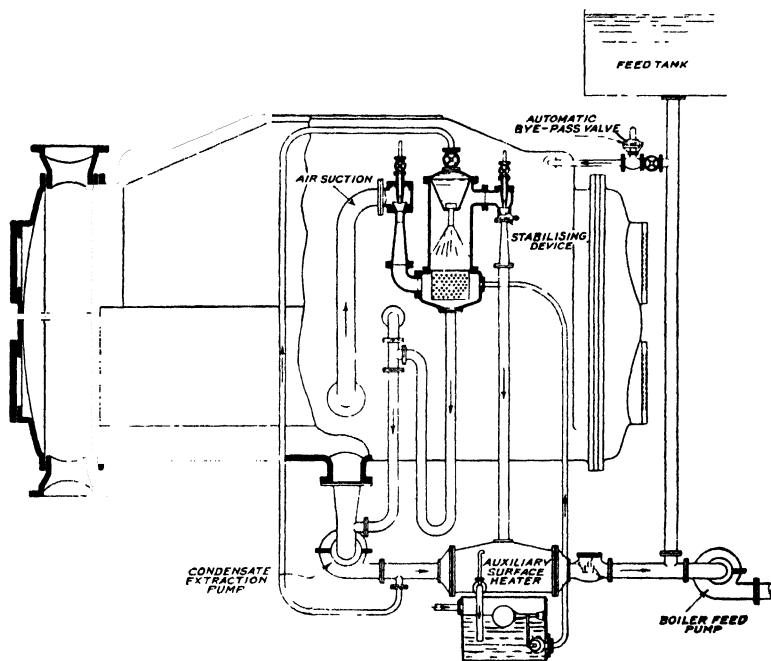


FIG. 140.

DIAGRAMMATIC ARRANGEMENT OF KINETIC EJECTORS.
(Contraflo Condenser Co., Ltd.)

to the suction of the condensate extraction pump as shown in Fig. 140. In the kinetic steam ejector, only one steam jet is used in each stage.

In the kinetic arrangement, as shown in Figs. 139 and 140, there are two special features—the stabilizing device used in the second stage ejector, and an automatic by-pass valve allowing condensate to be returned to the condenser.

Dealing first with the stabilizing device indicated in Fig. 140, and shown to a larger scale in Fig. 141, it will be seen from the latter

figure that there is a connection outside the combining tube of the second stage from the throat back to the inlet to this tube. When the stream in the throat is reduced, due to a reduced quantity of air to be extracted, air flows from the throat back to the inlet and so maintains the combining tube full. The sectional area of this by-pass can be controlled by a rotating shutter. The shutter is rotated by an eccentric pin on a spindle projecting to one side

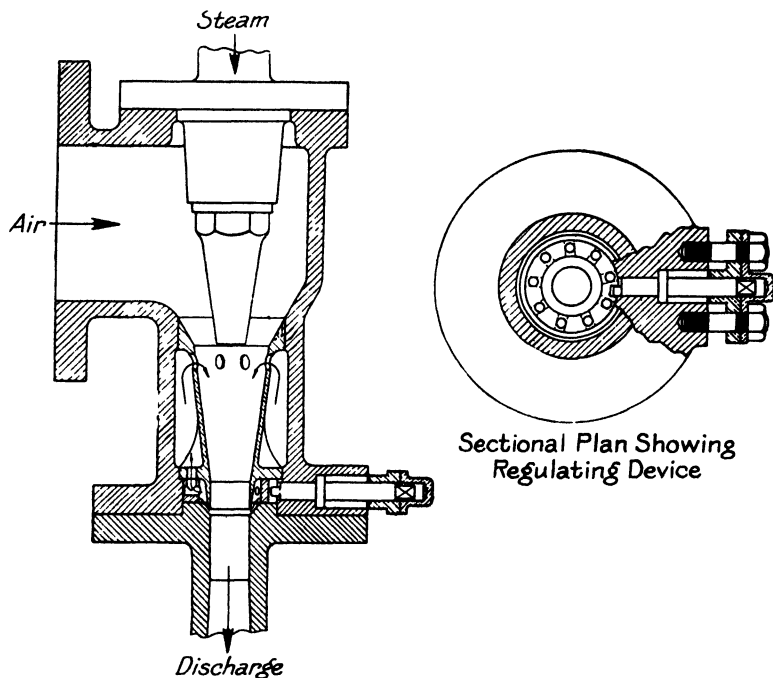


FIG. 141
STABILIZING DEVICE.
(Contraflo Condenser Co., Ltd.)

of the ejector. Once the correct setting has been obtained, as shown by the maintenance of vacuum with reduced air quantities, the spindle end can be covered by a cap and the device works automatically for given steam conditions.

The by-pass valve to which reference has been made is shown in detail in Fig. 142. In Fig. 137 it will be noticed in the upper right-hand corner. Dealing with the arrangement shown in Fig. 142, it will be seen that there is a flexible diaphragm on the valve spindle, above which is a spring, in tension when the valve is closed. The

upper side of the diaphragm is connected to a certain stage on the main turbine and the lower side to the condensate pump, discharge

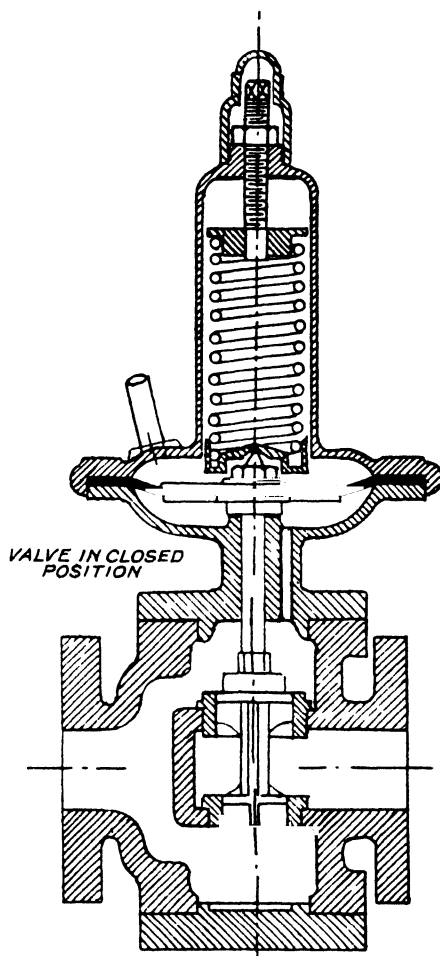


FIG. 142.
BY-PASS VALVE KINETIC SYSTEM.
(Contraflo Condenser Co. Ltd.)

side. The connection to the turbine and the spring tension are so selected that with the turbine carrying upwards of half load the spring tension is overcome and the valve is closed. At lower loads the spring opens the valve allowing water to be drawn into the

condenser, thus ensuring that the condensate pump has sufficient water to keep it operating satisfactorily

In Fig. 140 there will be seen a tank below the surface after-condenser into which the condensate from that condenser drains. Any make-up feed water required is also led into this tank, in which there is a ball float operated vacuum trap. As the float lifts the trap allows water to be drawn into the inter-condenser. Here the

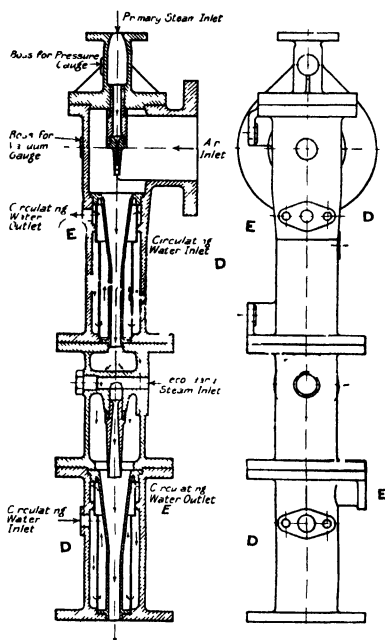


FIG. 143

DELAS SIMPLE EJECTOR

(Cole, Marchant & Morley, Ltd.)

water is de-aerated, the air passing to the second stage ejector and the water enters the feed water system and passes to the boiler. This trap is similar in principle to the trap illustrated in Fig. 135.

The Delas Ejector. In the Delas steam jet air extractor which is shown in Fig. 143, the method of operation is the same as in other two-stage ejectors but for the fact that the combining tubes in both stages are water cooled. The Delas system was originated by the Société des Condenseurs Delas, and ejectors of this type are made in England by Messrs. Cole, Marchant & Morley, and Messrs. The Brush Electrical Engineering Co., Ltd.

The steam supply and the single jets are similar to other types of steam jet extractor, but circulating water or condensate is passed round the combining tubes and tail pipes, entering at D and passing out at E, thus passing round both diffusing pipes in series. For

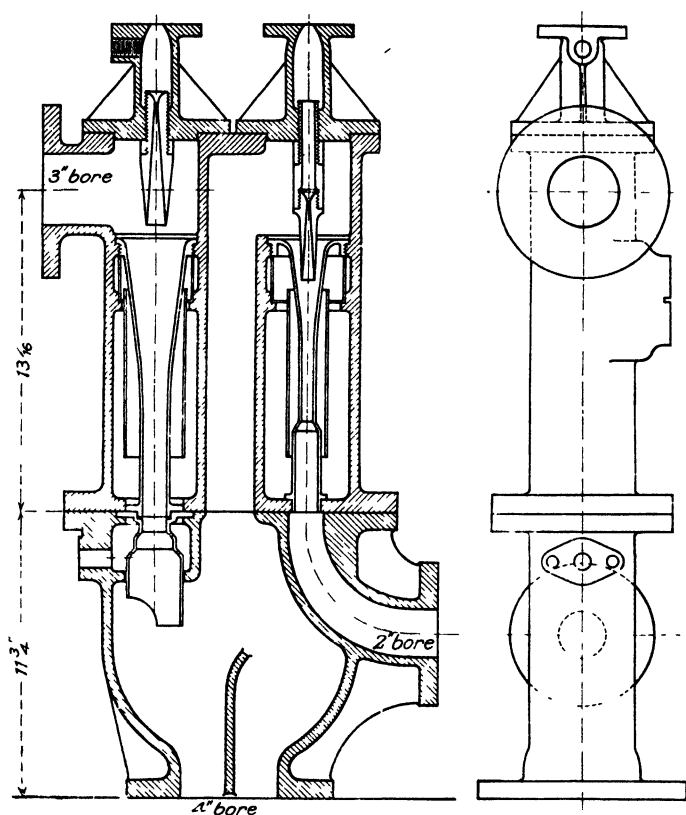


FIG. 144
ARRANGEMENT OF DELAS EJECTOR
(Cole, Marchant & Motley, Ltd.)

this system claims are made that it is more efficient than those with unjacketed extractors; these points are more fully discussed on pages 205–207.

When used with an inter-condenser (the great majority being of this type) the Delas ejector takes the form shown in Fig. 144. One striking feature of this apparatus is the compact arrangement due to the very small volume of the inter-condenser which is of the

jet type The injection water enters the first stage tail pipe by a few holes at the lower end, and this water passes away by the bottom flange D, the air rising between the two stages to the second stage entraining chamber

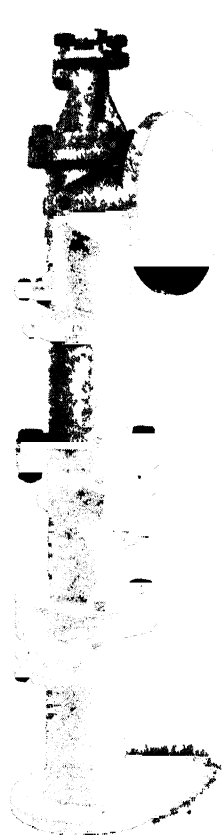


FIG 145
DIFFUSOR (SIMPLE)

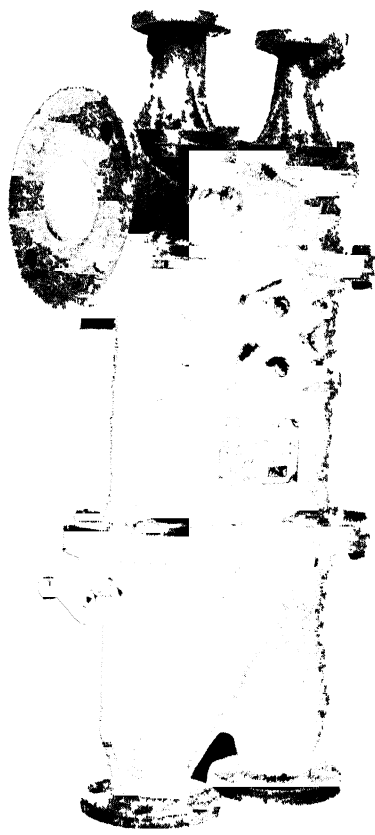


FIG 146
DIFFUSOR (INTER CONDENSER)
(Société des Constructeurs Français)

When water cooling such comparatively large tubes as are used for the diffuser and tail pipe, either a large volume of water or a small sectional area of passage has to be used to secure an adequate water velocity with the accompanying improvement in heat transmission. Small passages may become choked unless the water is carefully strained. It is most economical to use condensate.

External views of the Delas ejector of the simple and inter-condenser types are shown in Figs. 145 and 146.

The Ljungstrom Ejector. Some experimental work has been done in Sweden by the A. B. Ljungstroms Angturbin regarding steam jet air extractors. Some particulars have been published* of the simpler two-stage ejector used by Ljungstrom on his turbine-driven locomotive. A section through the ejector is shown in Fig. 147. The first stage is shown vertical, the steam supply to it

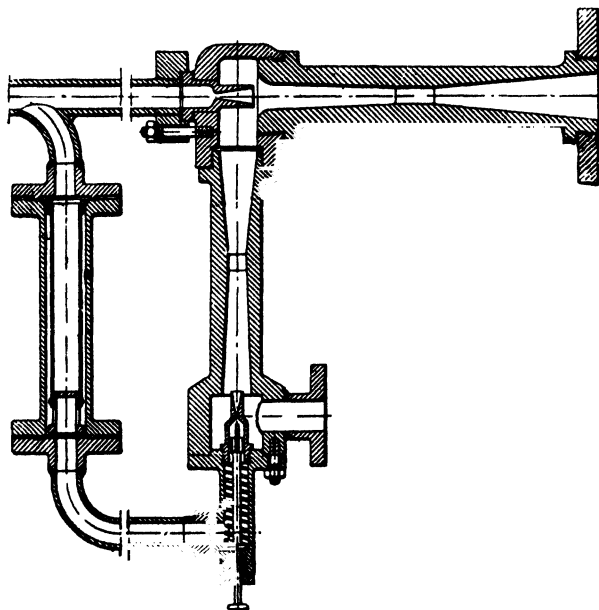


FIG. 147
STEAM EJECTOR.
(A. B. Ljungstroms Angturbin.)

being controlled by a needle valve. The proportions of the diffuser appear to be different to those adopted in other ejectors.

The Worthington Ejector. A steam jet ejector has been developed by Messrs. The Worthington Pump & Machinery Corporation, and also by Messrs. Worthington-Simpson, Ltd., and is illustrated in Fig. 148. From this it will be seen that this design only uses one jet in each stage, and that a surface inter-condenser is incorporated. Notwithstanding the use of the surface inter-condenser the discharge from the first stage has a fairly direct passage to the second stage.

* *Engineering*, 11th August, 1922.

The Ingersoll-Rand Ejector. A design of steam jet ejector which presents several departures from the usual design is that introduced by Messrs. The Ingersoll-Rand Co., Ltd. A section through the first stage ejector is shown in Fig. 149. The steam inlet is at the extreme left and it passes through an annular jet. Air entering by the lower flange passes to the diffuser both inside and outside the annular

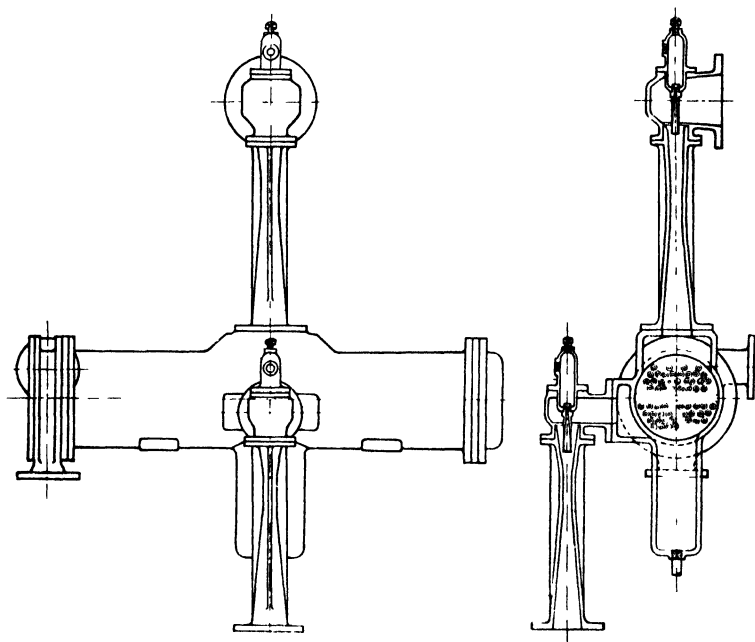


FIG. 148.

TWO-STAGE STEAM JET AIR EJECTOR WITH INTER-COOLER.

(Worthington Simpson, Ltd.)

steam jet. Externally the steam carries the air forward to the diffuser by surface friction, but internally the air must become mixed with the steam as the ring of steam will eventually become a cone. This is indicated by the shading in Fig. 149, showing the form which it is assumed the steam jet adopts.

For larger installations several of these ejectors may be used, and Fig. 150 shows four combined in one casing. In this figure, different views of the steam nozzles can be seen. The air inlets to the centre of the nozzle can be seen in three of the nozzles. The annular

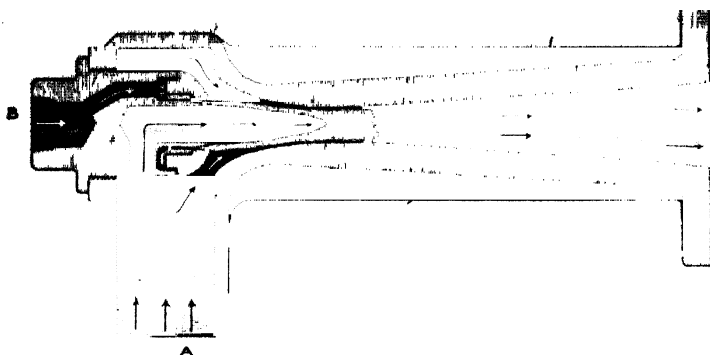


FIG. 149
STEAM INJECTOR—FIRST STAGE—SHOWING AIR AND STEAM
MIXING
(Ingersoll Rand Co. Ltd.)

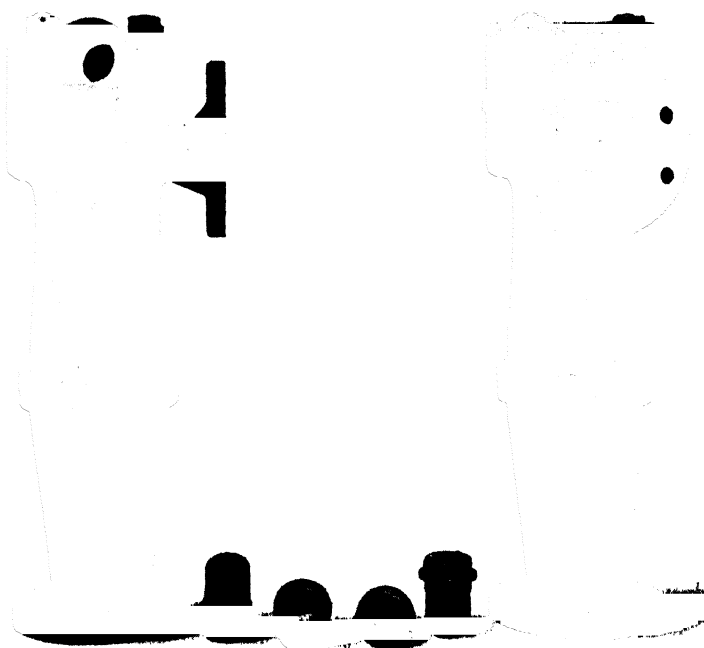


FIG. 150
BATTERY OF FOUR FIRST STAGE INJECTORS
(Ingersoll Rand Co., Ltd.)

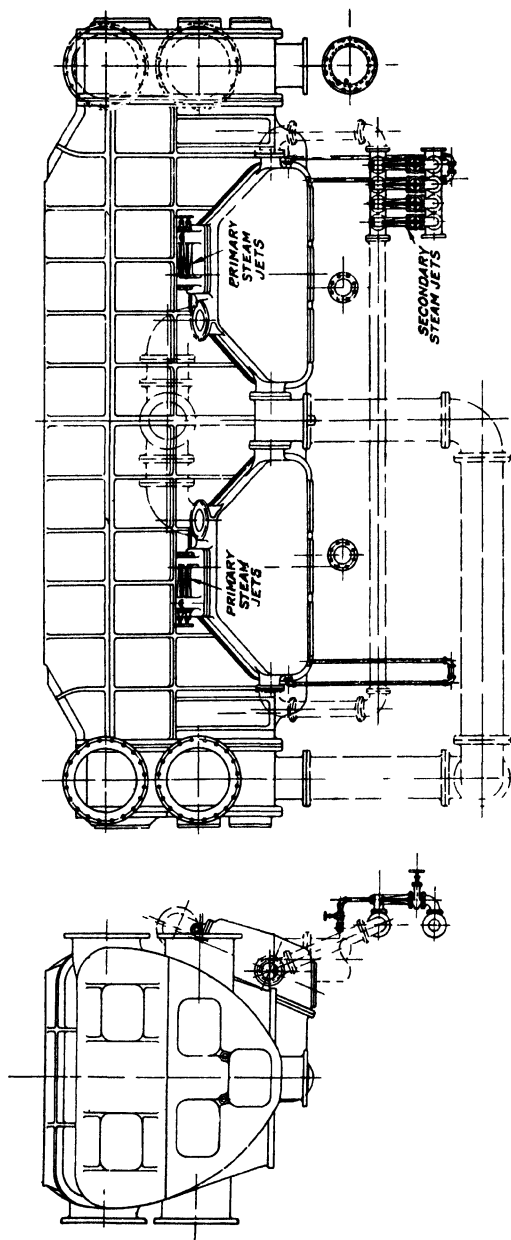


FIG 151
 ARRANGEMENT OF CONDENSER, AIR COOLER AND AIR EJECTORS
 (Ingersoll Rand Co., Ltd.)

steam nozzle is seen in the second from the left. The third from the left shows the steam inlet.

The arrangement of this ejector is shown in Fig. 39. Two ejectors are mounted on top of a cooler, described on page 91, and discharge to the right-hand side of the cooler. The discharge passes

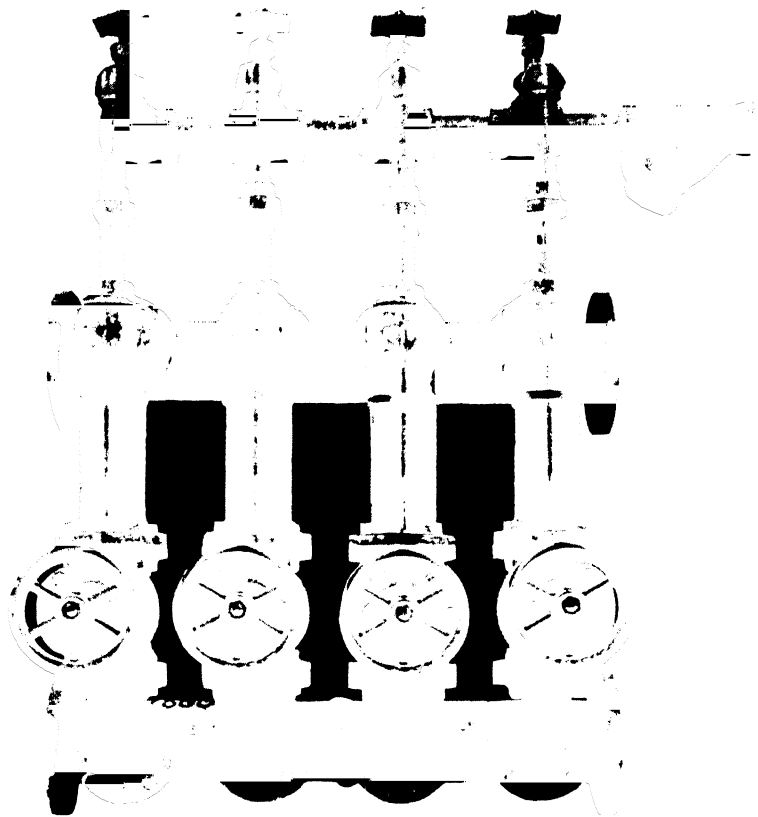


FIG. 152

BATTERY OF SECOND STAGE EJECTORS
(Ingersoll Rand Co., Ltd.)

through an inter-condenser which is a combined jet and surface condenser. A portion of the condensate is allowed to mingle with the exhaust and condenses some of the steam. A section through this inter-condenser is shown in Fig. 39*a*. It then leaves this inter-condenser by means of a vacuum trap, working on a principle similar to that illustrated in Fig. 134. The remainder of the cooling

is done by the circulating water passing through grids similar to those used in the air cooler section.

The complete arrangement is more clearly shown in Fig. 151. The air after passing through the inter-condensers which are at the outer ends of the two air coolers enter a common bus pipe from which it is extracted by secondary steam jets as shown. This battery of secondary steam jets is illustrated in Fig. 152, in which the steam supply enters through the strainer in the upper right-hand corner. The secondary ejectors are similar in design to the primary.

The Weir Ejector. The arrangement of steam jet ejector introduced by Messrs. G. & J. Weir is shown in Fig. 153. A surface inter-condenser is used, and the relative positions of the second stage

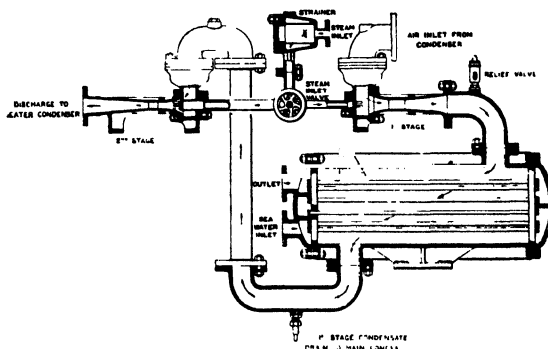


FIG. 153

STEAM EJECTOR ARRANGEMENT

(G. & J. Weir Ltd.)

ejector and the inter-condenser, etc., can be varied to suit special circumstances.

Brown-Boveri Ejector. Messrs. Brown-Boveri & Co., although using a hydraulic pump very largely, have designed a simple two-stage steam ejector as shown in Fig. 154, this presenting no unusual features.

Pipework Arrangement. A diagrammatic pipework arrangement of a condensing plant with steam jet air extractors is indicated in Fig. 155. This is designed for use in connection with house service turbines. The condensate is extracted by twin pumps, and a shunt on the discharge leads to the inter-condensers. Separate extraction pumps are used to withdraw the condensate from the inter-condensers and to discharge it back into the discharge from the main extraction pumps, thus completing the shunt. The condensate then passes to the house service set condenser.

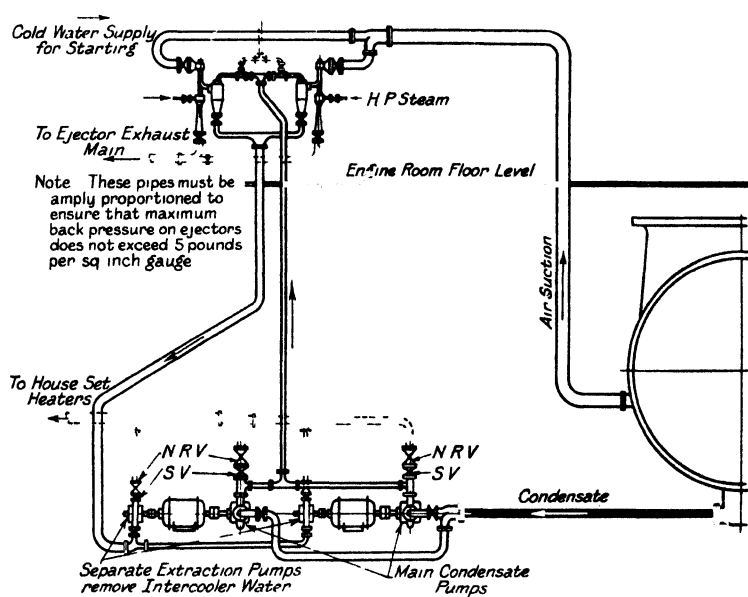
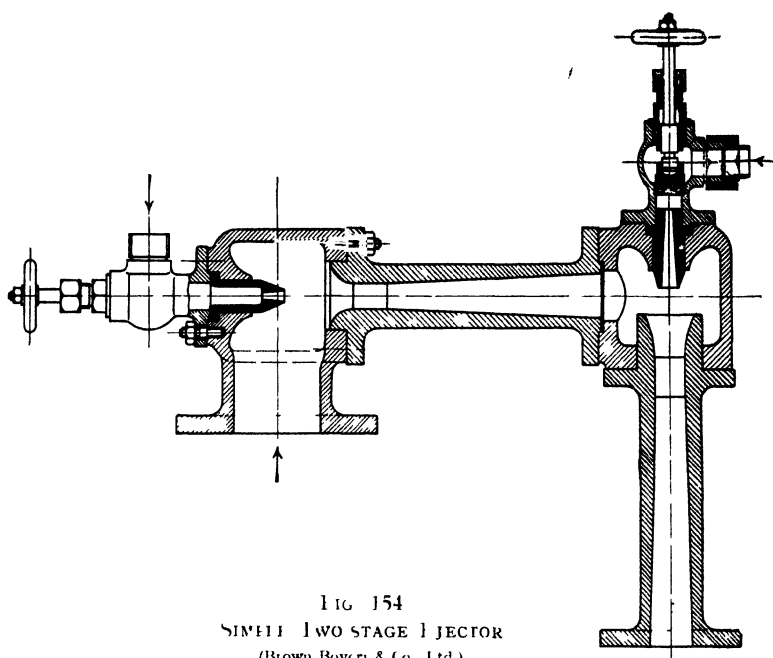


FIG. 155.

Meldrom Ejectors. Steam jets were used for moving air for a long time before they were used for creating a high vacuum. They are used, in the single-stage type for priming pumps, railway

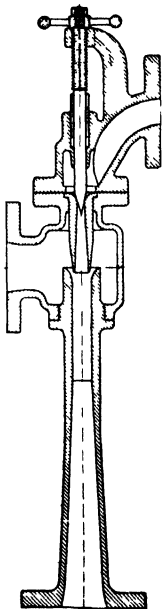


FIG. 156
MELDRUMS' STEAM EJECTOR

vacuum brakes, creosoting tanks, etc. A section through a single-stage ejector as made by Messrs. Meldrums, Ltd., of Manchester, for these and many other purposes is shown in Fig. 156.

CHAPTER XXII

HYDRAULIC AIR PUMPS

Leblanc Pump. The earliest rotary pump relying entirely upon hydraulic means for extracting the air was that introduced by M. Maurice Leblanc, and successfully developed by the Société Anonyme Westinghouse, of Paris and Havre.

The Leblanc Rotary pump, to the theory of which reference has been made on pages 213 to 219, has been manufactured for a long time in Great Britain by Messrs. The Mirrlees-Watson Co., Ltd., and the Metropolitan-Vickers Electrical Co., Ltd. A typical section through a modern example of this pump is shown in Fig. 157. The sealing water is led to the central drum *A*, whence it escapes by the nozzle *B*, passing through a rotating wheel *C*, which is in effect a reversed Pelton wheel. This wheel divides the water into a succession of small elements which are impelled down the diffuser *D* in the form of sheets. These sheets form a series of water pistons, and the air entering by flange *E* is trapped between successive sheets and carried out. Assuming that the sheet formation is maintained, the distance between the sheets decreases as the sheets pass down the diffuser, owing to the increase in pressure. The arrangement of the central drum *A* and the surrounding wheel *C* is clearly shown in the longitudinal section.

In order to start the pump working, water under the necessary pressure can be admitted to the central drum, or a steam ejector can be used.

When such a vacuum has been obtained that the operating water is aspirated naturally into the drum, the ejector can be closed down.

It will be seen that the Leblanc pump is a positive displacement pump, in which water pistons are used.

On these general lines, a large number of these pumps have been made by the firms named and by others in other countries. Variations have been made in the angles and outlines of the blades in the wheel *C* and in the inclination of the diffuser *D*, with the object of getting more stable operations with larger quantities of air.

It was always found that, with air quantities even only moderately in excess of the amount for which the pump was designed, that the vacuum fell rapidly and the pump became unstable in operation.

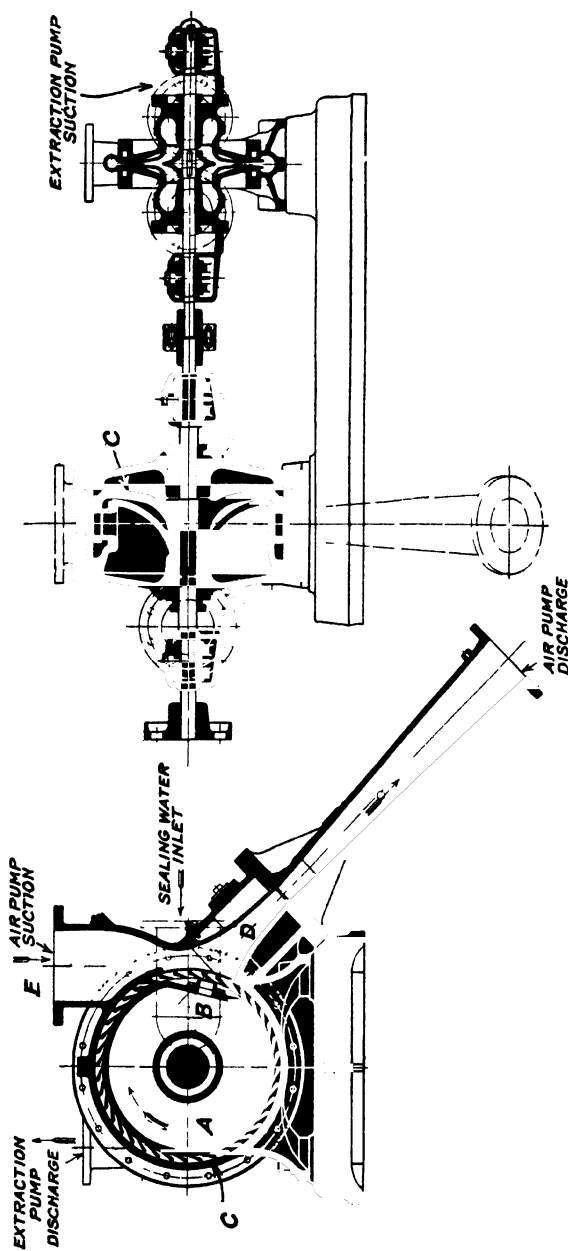


FIG. 157.
LEBLANC ROTARY AIR PUMP.

Assuming that the water level in the sealing tank is, say, 6 ft above the pump centre line, then the total head under which the water flows through nozzle *B* is this head plus the head corresponding to the vacuum. If the vacuum is 28 ins the total head at the nozzle is 37.7 ft and the velocity corresponding to this would be 49.27 ft per second neglecting for comparative purposes the inefficiency of the nozzle or the *vena contracta*. If the air inflow be increased so that the vacuum is reduced to 24 ins, the head becomes 33.2 ft and the velocity 46.24 ft per second. The water

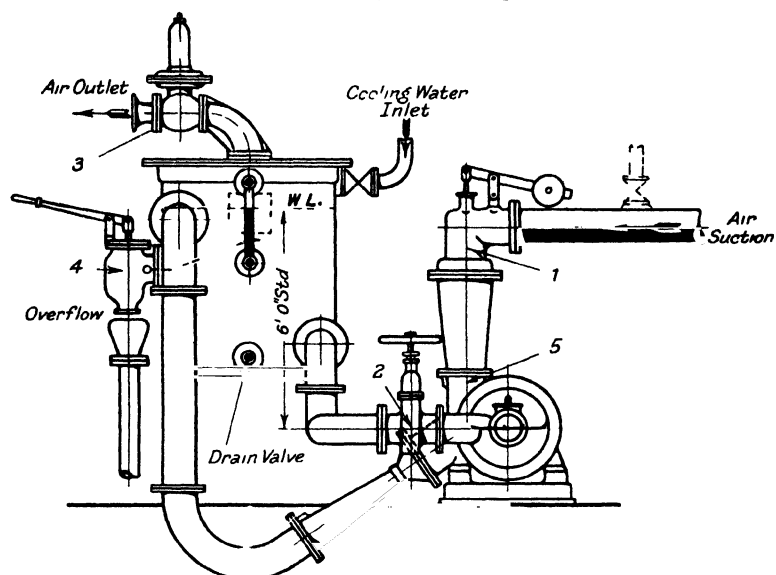


FIG. 158

LEBLANC PUMP SCANES CLOSED SEALING TANK

velocity and the quantity of water are each reduced about 6 per cent, and the energy in the water which depends on mass $\times$ velocity squared is reduced in the ratio of $(0.94)^3$ or about 0.83, i.e. a reduction of 17 per cent. The effect of the greater inflow of air is thus cumulative each drop in vacuum reducing the water quantity which in turn reduces the air abstracting capacity.

THE SCANES PRESSURE TANK This characteristic has been overcome in a simple manner by the Scanes pressure tank. The arrangement is shown in Fig. 158. The sealing tank is elevated above the centre line of the pump and is closed. The sealing water, after passing through the diffuser, is returned to the sealing tank, taking

with it all the air extracted from the condenser. The only outlet for the air when liberated at the surface is through a special regulating valve (3) in Fig. 158, and shown in section in Fig. 159. The object of this valve is to retain the air in the tank when the vacuum

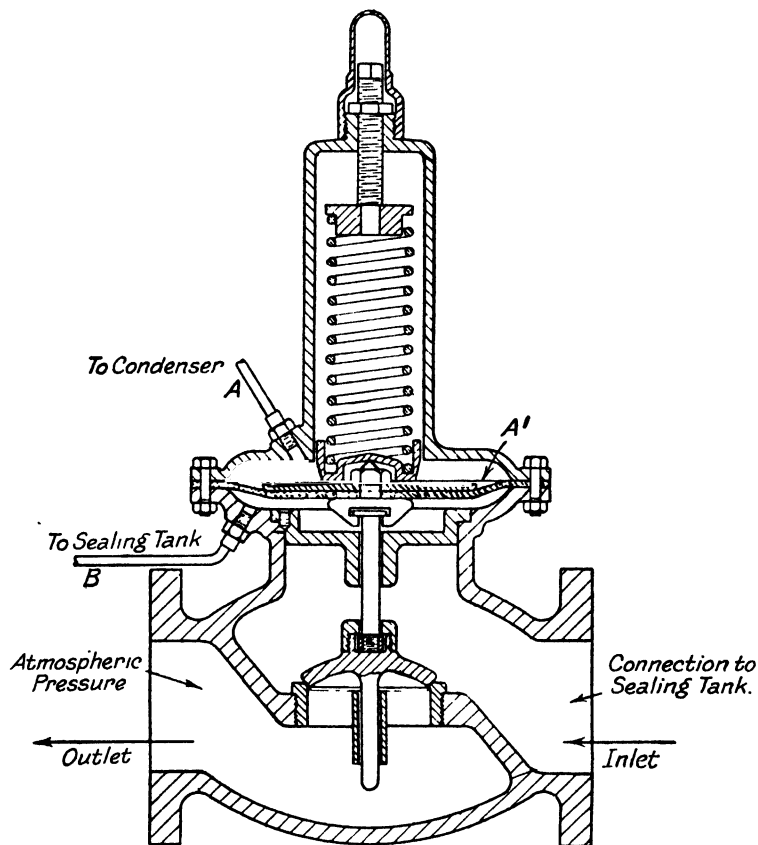


FIG. 159.
SCAN'S CLOSED TANK AIR CONTROL VALVE.

falls and thus, by putting pressure on to the water, to compensate for the diminished suction head. By this means the total head is kept constant and so the quantity of water passing through the nozzle remains unchanged. In this valve there is a diaphragm A, the upper side of which is connected to the condenser vacuum space and the under side to the sealing tank. Under normal working conditions the diaphragm is in its highest position and the attached

valve open, allowing free escape of the air from the tank. With a fall in vacuum, the increased pressure above the diaphragm plus the spring pressure tend to close the valve and thus throttle the air. This throttling is increased until the increase in air pressure in the tank is equal to the increase in absolute pressure in the condenser. The effect of this pressure tank will be shown later.

As will be seen from Fig. 158, a non-return valve is fitted to the

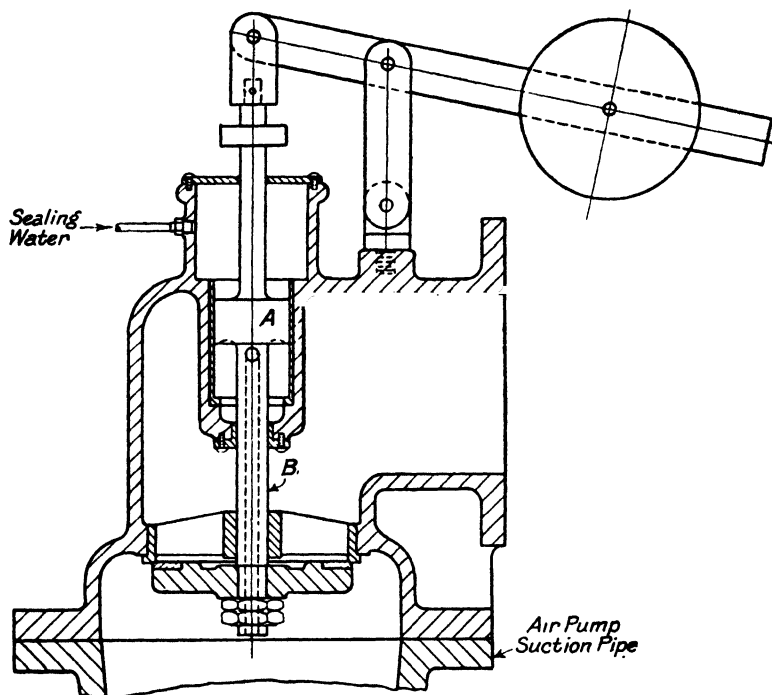


FIG. 160
SCANES CLOSED TANK, NON-RETURN VALVE.

air pump suction, to prevent water flowing back into the condenser should the pumps be stopped while there is vacuum in the condenser. A section of this valve is shown in Fig. 160. It will be seen that on the main spindle of the valve there is a small piston. The underside of this piston is connected by a hollow spindle to the air pump side of the valve. The upper side is water-sealed and subjected to atmospheric pressure. If the water rises in the pump and the vacuum fails, the valve is closed by the balance weight dropping. This valve is claimed to be sensitive and positive.

It is important that the water supplied to the pump should be as air free as possible and thus a large water surface is desirable for air

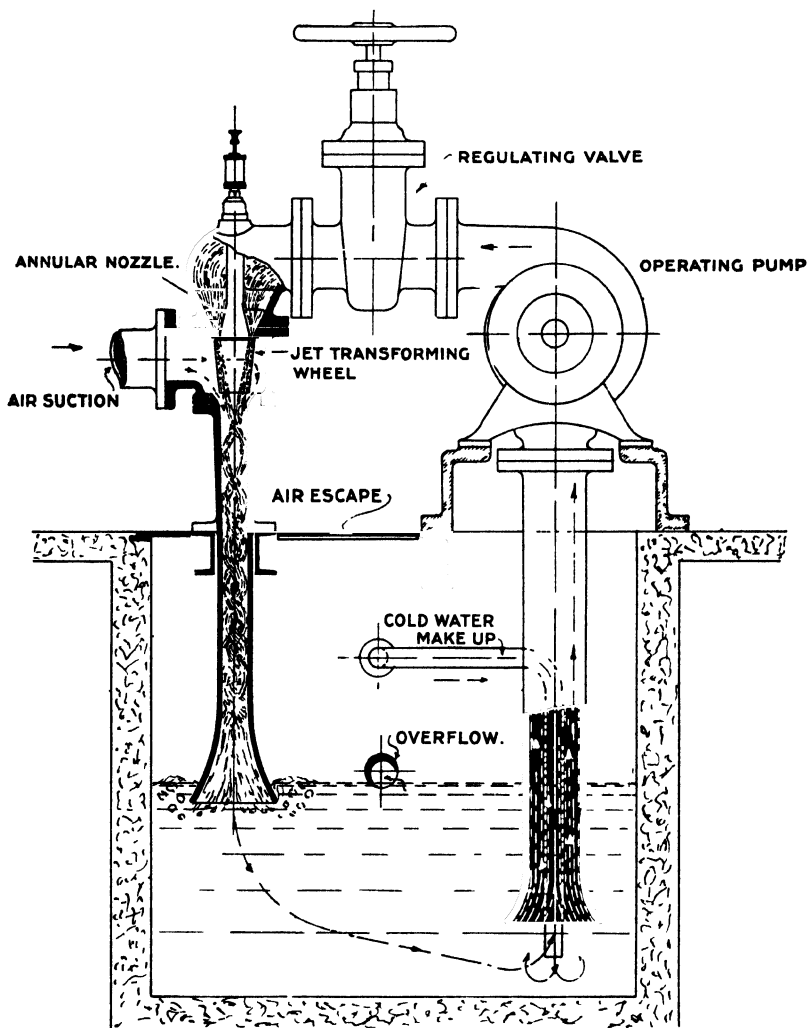


FIG 161.

WORTHINGTON-SIMPSON ROTARY AIR EJECTOR ARRANGEMENT

liberation. In Fig. 158, this has been obtained in a cylindrical tank of comparatively small dimensions by so arranging the water inlet

and outlet tangentially that the water forms a vortex. The surface is thus considerably increased over the still surface area.

On account of the condensation of vapour carried away by the air and of the energy dissipated in the pump, the sealing tank water will gradually rise in temperature and a supply of cooling water has to pass through the tank. The connection for this is indicated in Fig. 158. There is a float controlled overflow leading into an open bell-mouth drain pipe.

The Worthington Hydraulic Pump. Another form of hydraulic pump in which water pistons are used, is that adopted by Messrs. Worthington-Simpson, Ltd., and the Worthington Pump & Machinery Corporation. The general arrangement of the plant is shown in Fig. 161, from which it will be seen that it consists mainly of a centrifugal pump supplying operating water under pressure and of the hydraulic vacuum pump itself. Fig. 161 shows the plant arranged over a concrete tank in the floor of a power station, but it can be arranged as a self-contained plant with a separate sealing tank.

The main portions of the air extractor is shown in Fig. 162. The operating, or "hurling" water, to adopt the name used by the makers, enters by the upper right-hand flange and passes between the nozzle rings in the base of the chamber. The hurling water then impinges on to a jet transforming wheel which is attached to the lower end of a vertical spindle, free to rotate. In the outer portion of this wheel divergent nozzles have been formed, inclined about 15° to the axis of the wheel. This wheel and a development of the outer surface of it are shown in Fig. 162. The water passes from the upper chamber with a motion practically axial, and coming upon the nozzles inclined at 15° , causes the wheel to rotate. The water leaves the wheel with a rotary motion, and instead of a series of jets moving towards the throat of the tail pipe with a

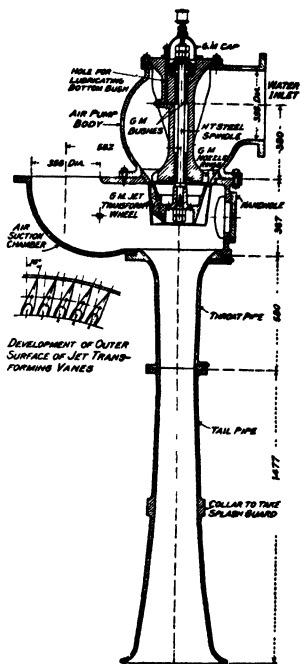


FIG. 162.

SECTIONAL ARRANGEMENT
OF HYDRAULIC AIR
PUMP

(Worthington-Simpson, Ltd.)

linear motion we have the jets moving with an imposed rotary motion. The action of the water in carrying out the air, which enters by the upper flange on the left hand, is probably due to two causes. In the first place, each jet of water issuing from the wheel with combined rotary and axial motion may form successive sheets of water passing into the throat pipe and down the tail pipe in a spiral form. Such complete sheets, if formed, would act as water pistons and carry air out between them. But owing to the splashing that must occur under the jet transforming wheel, the formation of complete sheets is very doubtful, and it is more probable that

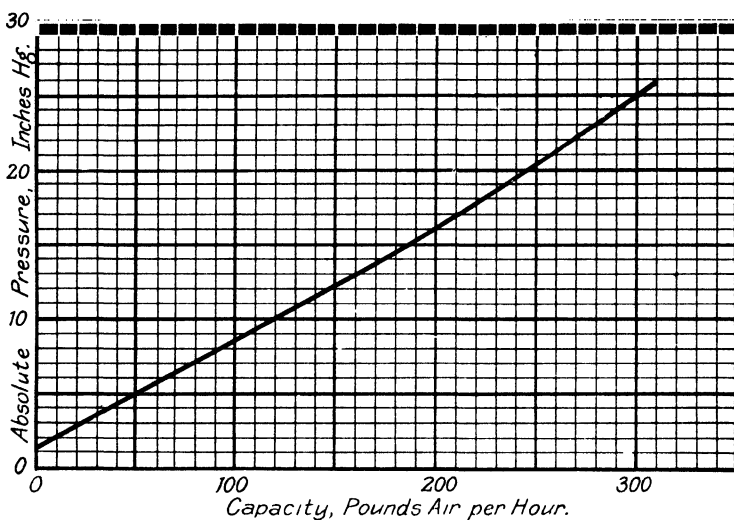


FIG. 163.

CHARACTERISTIC OF WORTHINGTON PUMP.

the air is carried along with the rapidly moving water by surface friction. The air thus becomes entrained in the surface of the moving jets or the drops or other sub-divisions into which the jet disintegrates.

If the pump is equivalent to a positive displacement pump, then with hurling water at a constant temperature, it would follow that the air withdrawal capacity of a particular pump would be proportional to the air pressure at the inlet to the pump. That this is the case with the Worthington pump appears to be indicated by Fig. 163, in which it will be seen that the connection between capacity and absolute pressure at the inlet is linear up to an absolute pressure of 15 ins. Hg. The departure from a straight line is not

great for higher absolute pressures. Thus the claim that the pump is equivalent to a positive displacement pump appears to be well supported.

Fig. 164 shows a typical arrangement of condensing plant with the Worthington hydraulic vacuum pump and rotary two-stage condensate extraction pump. A separate tank is provided into which the tail pipe of the air pump discharges under water level. To keep the operating water at a constant temperature, there is a connection

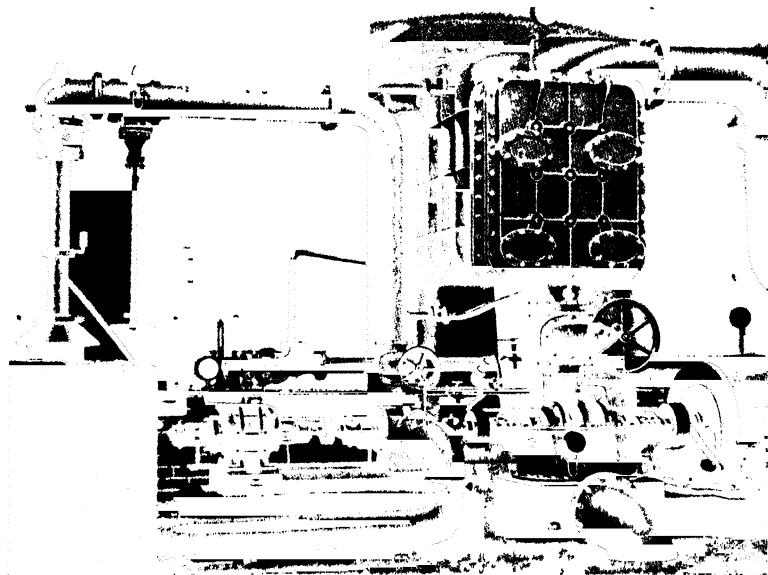


FIG 165

TYPICAL SMALL PLANT (2,000 kw)
(Worthington Simpson, Ltd)

from the circulating water inlet to the tank and from the operating pump discharge back to the circulating water outlet. The balance connection to the condenser for the extraction pump should be noted.

This arrangement of keeping the water in the sealing tank at a constant temperature by connections to the circulating water supply, rejects from the system the heat contained in the steam carried from the condenser with the air.

A representative small plant (2,000 kw) with this type of pump is shown in Fig. 165.

Escher Wyss Hydraulic Extractor. Another arrangement of an hydraulic air extractor has been adopted by Messrs. Escher Wyss & Co., of Zurich, this being shown in Fig. 166. As in the Worthington

extractor, an ordinary centrifugal pump supplies water under pressure to the ejector. This consists of one or more annular nozzles *A*, through which the water passes into the discharge pipe *B*. The opening of the nozzles can be controlled by the handwheel *C*.

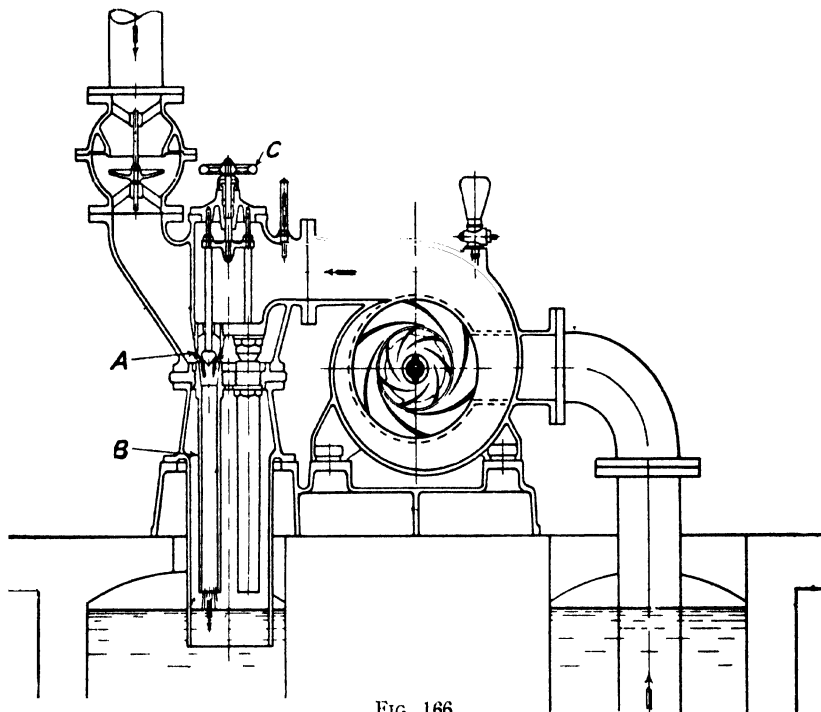


FIG 166.

ARRANGEMENT OF HYDRAULIC AIR PUMP

(L. Scher Wyss & Co)

Müller Air Extractor. Another type of hydraulic air extractor is that introduced by Dr P. H. Müller, of Hanover, and made in England by Messrs. Willans & Robinson, Ltd (now incorporated in the English Electric Co., Ltd), and also by Messrs. Daniel Adamson & Co., Ltd

In Germany this type of pump is adopted by many of the leading constructors of condensing plant, amongst them being the following :

Maschinenfabrik Augsburg Nürnberg, of Nürnberg
 Frederick Krupp (A G), of Kiel
 Balcke (A G), of Bochum
 Dortmunder Vulkan, Dortmund
 Maschinenfabrik Grevenbroich
 Schuchtermann & Kremer, Dortmund
 Otto Estner, Dortmund

This pump is also made in France by the Société Anonyme de Constructions Electrique, of Mulhouse, and La Companie Française Thomson Houston. It is also made in Belgium, so that it will be

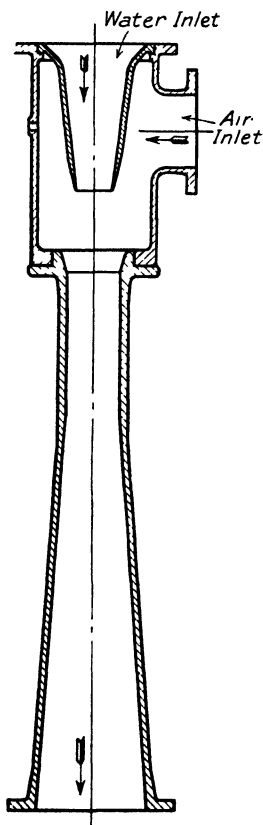


FIG. 167.
MÜLLER WATER EJECTOR
BY GREVENBROICH.

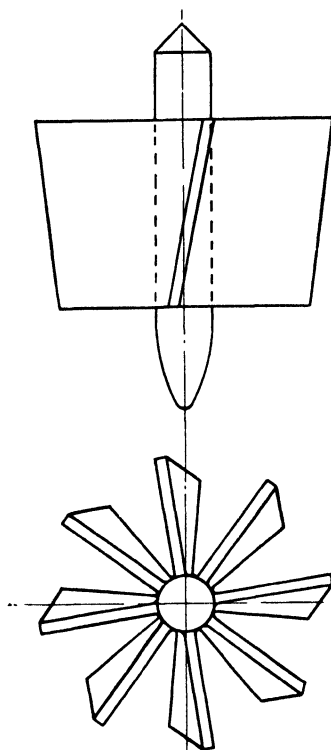


FIG. 168.
MÜLLER WATER SPINNER.

seen that it has been adopted very largely by Continental manufacturers.

The Müller pump is essentially a water ejector in which the operating water is obtained from the circulating pump, or may consist of the entire circulating water. It is worked in three different ways which have been termed the series, shunt, and separate pump methods. In the series method, now rarely used, the whole of the circulating water first passes through the ejector

and then through the condenser as usual. In the shunt method, the necessary quantity of water is diverted on the discharge side of the circulating pump, and passing through the ejector is returned to the circulating pump suction. In the former type, the head on the circulating water is increased by the amount required in the ejector, whereas in the latter the head supplied to the ejector is the same as that required for the condenser. Should neither of these arrangements be convenient nor economical for a particular case, resort is made to the use of a separate pump which draws from and discharges back to the circulating water supply via the ejector.

It should be noted that with the air extractor in series with the condenser, the Müller ejector has to discharge against a considerable head, and under these conditions operates satisfactorily, whereas some other hydraulic pumps are only satisfactory if the discharge head does not exceed 1 lb. per sq. in. above the atmosphere.

Fig. 167 shows a section through a Müller ejector as made by the Grevenbroich Company. The water enters by the top flange and the air by the flange on the right-hand side. In the upper part of the nozzle, there is placed a scroll which gives a spin to the water. This scroll is shown separately in Fig. 168. It consists of a star wheel with eight radial vanes each inclined to the axis of the wheel. By its action, the scroll divides the water up into a number of spiral sheets which form water pistons and entrap the air between them as they pass from the nozzle to the diffuser entrance. With this formation, the pump is equivalent to an actual displacement pump.

The scroll shown in Fig. 168 is about $6\frac{1}{2}$ ins. diameter at the lower end and about $4\frac{3}{8}$ ins. long. The inclination of the vanes is $\tan^{-1} 0.168$, or about $9^{\circ} 30'$. The complete ejector was designed to pass 2,028 gallons per minute, the external head being 34 ft. 6 ins. and the vacuum 27.6 ins. Under these conditions the water velocity through the spinner at its lower end is about 26 ft. per second.

A diagrammatic arrangement of the shunt system is shown in Fig. 169. This shows the motor-driven circulating pump P, with a branch on the discharge B leading to the Müller ejector. Air leaves the condenser K by pipe L, and passing through a non-return valve, enters the ejector by the side flange. The aerated water returns by pipe F to the circulating water supply canal or main. There is also indicated the condensate extraction pump, etc.

The next figure, No. 170, shows the arrangement with a separate pump, the special pump P_1 getting its supply from the suction side of the main circulating pump P.

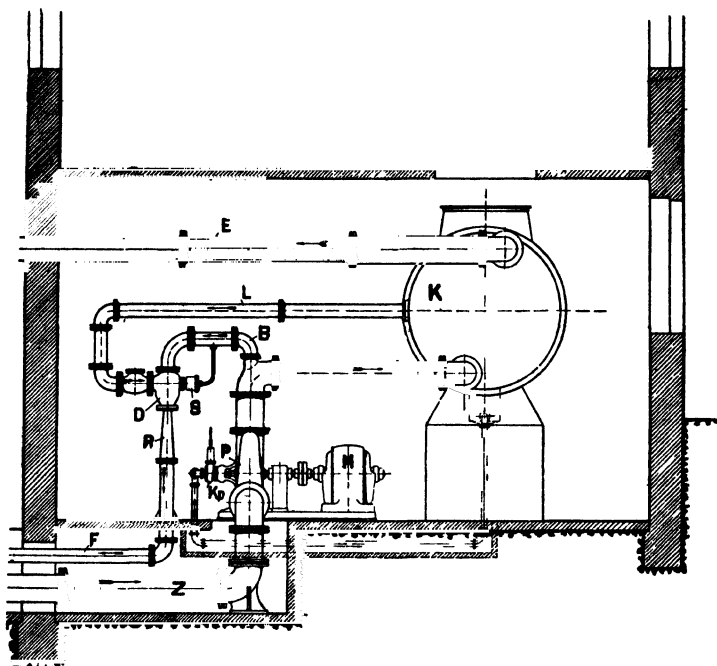


FIG 169

MÜLLER SHUNT SYSTEM

(Maschinenfabrik Augsburg Nürnberg)

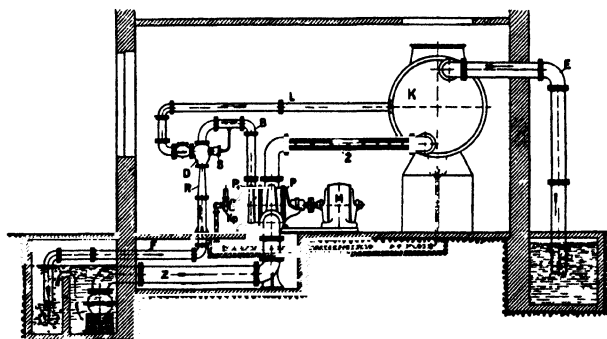


FIG 170

SPECIAL PUMP MÜLLER SEPARATE PUMP SYSTEM

(Maschinenfabrik Augsburg Nürnberg)

In the series method, the air ejector initially was incorporated in the lower part of the condenser as indicated in Fig. 171, but was usually external to the condenser as in Fig. 172. In the latter, there

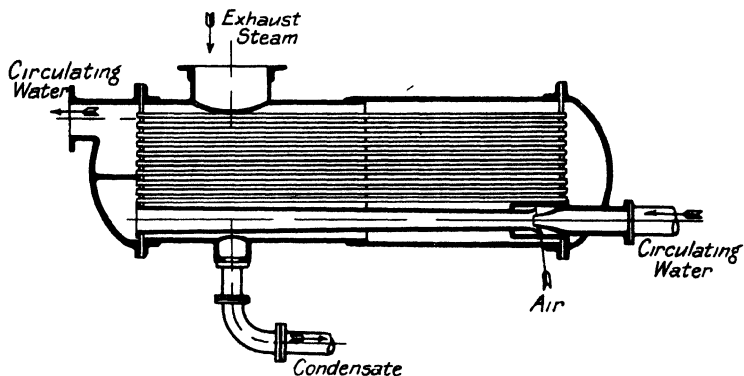


FIG 171

COMBINED CONDENSER AND MÜLLER EJECTOR

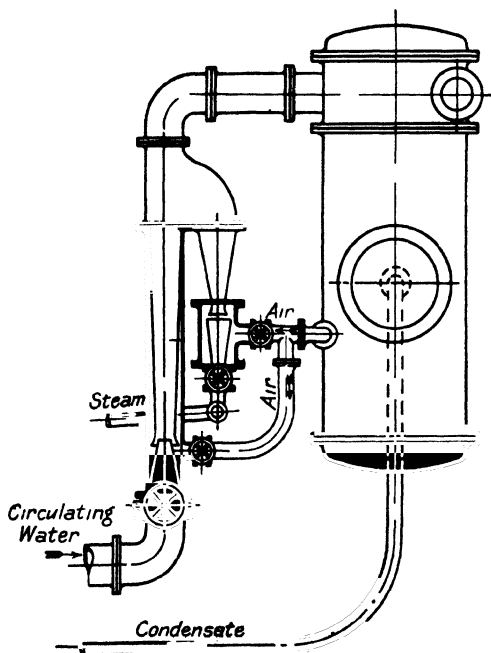


FIG 172

MÜLLER SERIES EJECTOR WITH STEAM EJECTOR.

has been provided a steam-jet ejector using exhaust steam from the turbine driving the circulating pump, the steam ejector working in parallel with the water ejector. This combination has the great drawback that the whole heat contained in the exhaust steam is lost, being carried away in the circulating water. In the series arrangement, the water ejector can be placed after the condenser.

MEANS OF PREVENTING FLOODING. In the Müller pump it is essential to guard against the possibility of the operating water

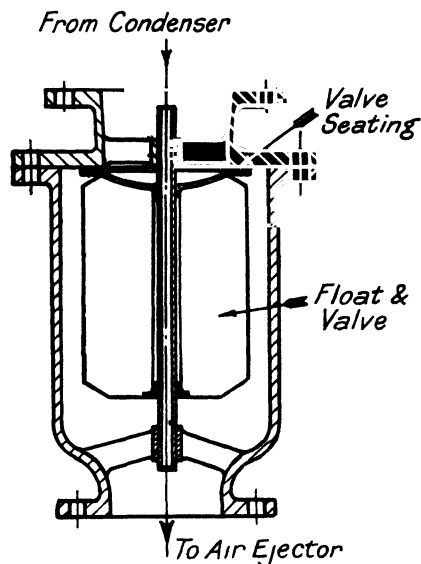


FIG. 173
NON-RETURN VALVE FOR MÜLLER EJECTOR
(Dortmunder Vulkan)

flooding into the condenser should the pump fail, and non-return valves are fitted for this purpose. Fig. 173 shows a float valve used for this purpose by the Dortmunder Vulkan, but the flap valve illustrated in Fig. 174 is generally used. In this valve, as made by Daniel Adamson & Co., Ltd., the hinged valve is made of aluminium, and it is kept open by the passage of the air and closes should the withdrawal of the air cease. Being light it does not cause much resistance to the passage of the air from the condenser.

In addition to this non-return valve, a vacuum breaking valve is fitted by which air is admitted to the air pump suction as soon as the pressure of the operating water falls below a predetermined

valve. This valve is shown in Fig. 175. When the pump is standing the piston shown is held in its lowest position by a spring, but

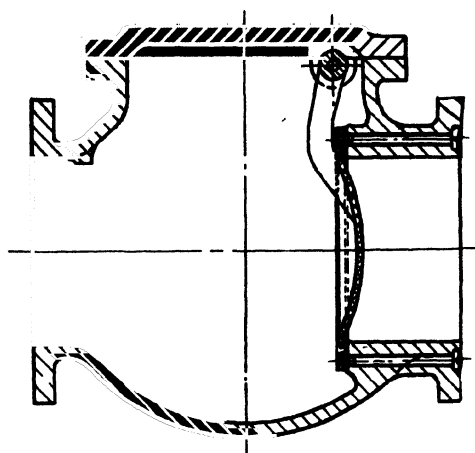


FIG. 174
NON-RETURN VALVE FOR MÜLLER EJECTOR.
(Daniel Adamson & Co., Ltd.)

it is raised as the water pressure is increased as the pump discharge is connected to the underside of the piston. This closes the atmospheric connection, the action being reversed when the pump is stopped either accidentally or deliberately. The vacuum is thus

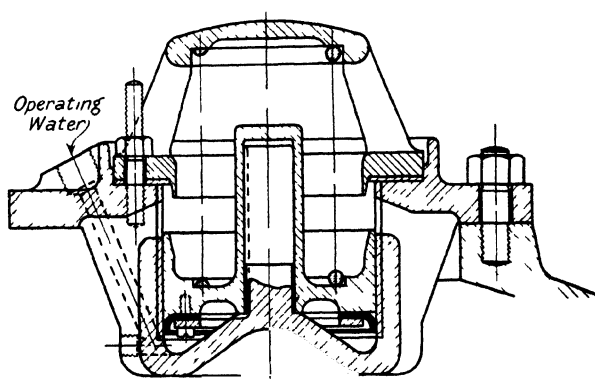


FIG. 175
VACUUM BREAKING VALVE FOR MÜLLER EJECTOR.
(Daniel Adamson & Co., Ltd.)

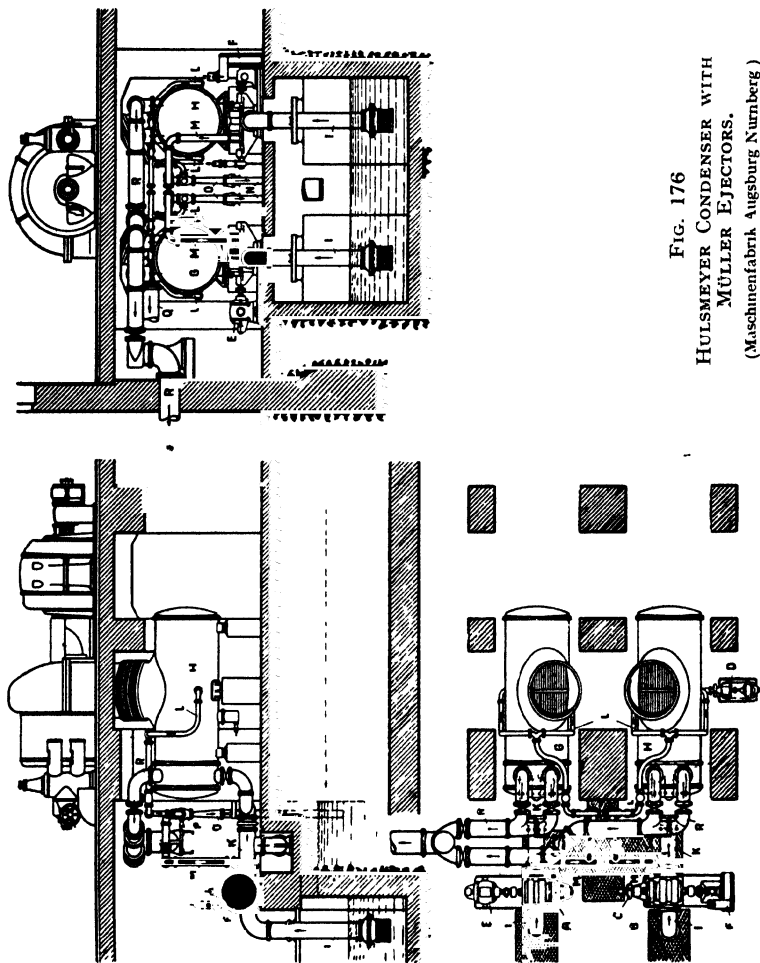


FIG. 176
 HULSMAYER CONDENSER WITH
 MÜLLER EJECTORS.
 (Maschinenfabrik Augsburg Nürnberg)

broken before the pump actually stops, and water is prevented from entering the condenser

MULLER EJECTORS WITH HULSMEYER SYSTEM CONDENSERS An arrangement of a large condensing plant with Muller ejectors is shown in Fig 176 This plant has twin condensers on the Hulsmeyer system, of which a description is given on page 119 Duplicate pumps are provided each supplying one condenser One set of pumps is driven by a motor and the other by a steam turbine Separate pumps are provided in each unit for the ejector water Fig 177 illustrates the pumping units in this particular plant,



FIG 177

ARRANGEMENT OF AUXILIARIES FOR HULSMEYER SYSTEM

(Maschinenfabrik Augsburg Nürnberg)

which was constructed by the Maschinenfabrik Augsburg Nürnberg, of Nürnberg

Fig 178 shows the arrangement of a series plant by Daniel Adamson In this it will be noticed that the scroll is shown in position

REJECTION OF AIR AND LOSS OF CONDENSATE In common with other hydraulic ejectors which do not use condensate as operating water, this ejector has the great advantage of discharging the air into the circulating water and so ejecting it entirely from the steam cycle of the system As every effort is made to reduce air leakages into the system to a minimum, and (see Chapter XXVI) as even

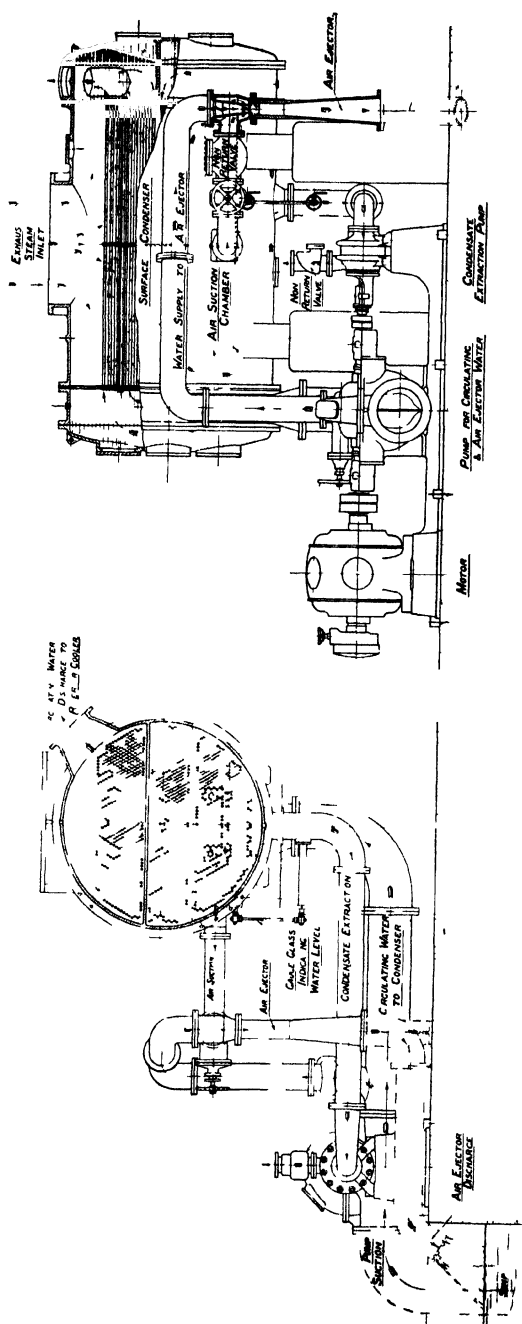


FIG 178
ARRANGEMENT OF CONDENSING PLANT BY MESSRS DANIEL ADAMSON

special apparatus is installed in large plants to abstract the air from the feed water to the last possible degree, it does not seem correct to use the condensate in such a way that it becomes saturated with air.

On the other hand, this type of hydraulic extractor carries away to the circulating water the vapour contained in the air as it leaves the condenser. Thus, as shown on page 199, in a plant maintaining a vacuum of 28 ins. with circulating water at 80° each pound of air

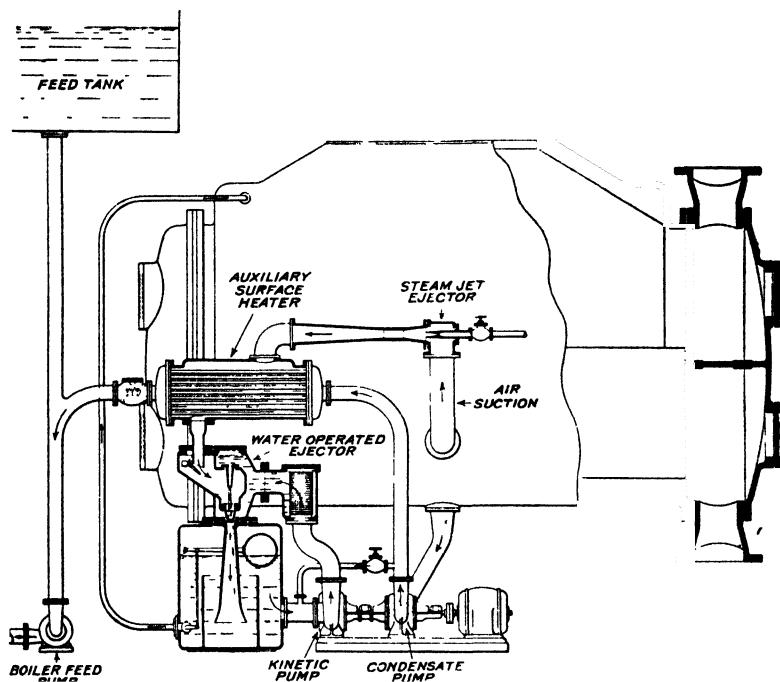


FIG. 179.

ARRANGEMENT OF KINETIC PLANT.

(Contraflo Condenser Co.)

removed, if at a temperature of 85° F. would carry practically an equal weight of vapour, and if the temperature be as high as 95° (see tests detailed on page 368) there would be over three pounds of vapour per pound of air. Bearing in mind the comparatively small weight of air usually present, say 40 to 50 lbs. per 100,000 lbs. of steam (and these figures are greatly in excess of good practice), even the higher figure is not excessive corresponding at the most to 0.15 per cent of the total condensate. The figures given on page 199 assume that there is no priming, but if this occurs, the loss of

condensate may be appreciable and have some effect upon the measured overall consumption of the prime mover.

With a modern plant, priming is probably not very serious and the loss of condensate immaterial. If the air suction temperature is high the hydraulic ejector is in effect a very small jet condenser working in series with the main surface condenser, but any such

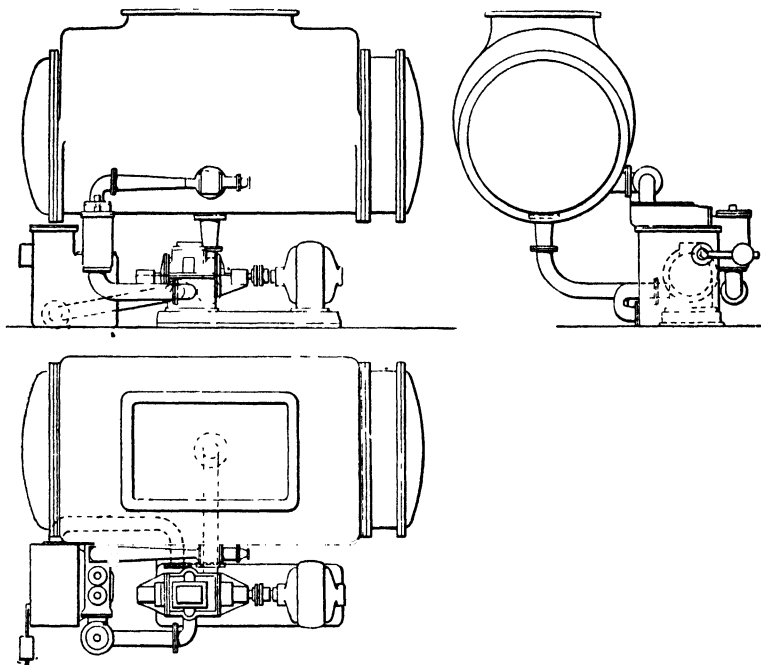


FIG. 180.

KINETIC PLANT ARRANGEMENT.
(Contraflo Condenser Co.)

effect can be traced by checking the temperature rise of the ejector water.

The Kinetic Rotary Pump. The Kinetic rotary pump is classified under hydraulic pumps, although it is not exclusively hydraulic, as the first stage of the air extraction is done by means of a steam ejector. The arrangement is shown in Fig. 179. As indicated, the plant is arranged for a closed circuit or a "closed feed" system, of which more will be said later, but this does not affect the principle of the kinetic rotary pump. In this, the first stage steam ejector is seen discharging into a surface inter-condenser. Through this

condenser the discharge from the condensate pump is passed on its way to the boiler feed pump suction.

The air with its contained vapour, passes from the inter-condenser to the entraining chamber of the second stage or water ejector. This water ejector consists of a central jet of water and an annular jet which enters at, or just previous to, the throat of the diffuser. The air is thus discharged into the sealing tank whence the water is abstracted by the kinetic pump and passed round the water ejector again. Due to condensation the water level in the tank would gradually rise and a float-controlled valve is provided, allowing the surplus water to be passed to the condenser, thus entering the feed water system.

When the closed feed system is not used, the inter-condenser can be omitted and the steam ejector discharges direct into the water ejector entraining chamber. Thus the water from the kinetic pump condenses the whole of the first-stage jet steam as well as completing the extraction of the air. To conserve the heat of this steam and to prevent the water in the sealing tank becoming too hot, the condensate pump then discharges into the tank and the overflow from the tank passes to the hot well. This arrangement is used extensively and is shown diagrammatically in Fig. 180. The condensate extraction and kinetic pumps are combined in one casing, and a twin water ejector is shown. The kinetic pump discharge passes through a strainer before reaching the water jet, this strainer also being shown in Fig. 179.

The Brown-Boveri Hydraulic Ejector. Another type of

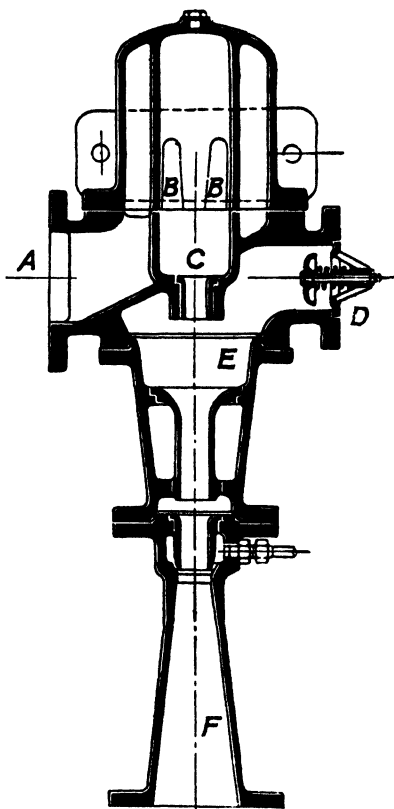


FIG. 181
HYDRAULIC EJECTOR.
(Brown Boveri, A. G.)

hydraulic ejector has been developed by Messrs. Brown, Boveri & Co., of Baden, Switzerland. The ejector is illustrated in Fig. 181. The operating water under the requisite pressure enters by flange A, and passing through the ports B, reaches the nozzle C. The ejector is connected to the condenser by flange D, in which is fitted a non-return valve to prevent the water entering the condenser should the plant be stopped with vacuum in the condenser. In the entraining chamber E, the air is picked up by the water jet and carried out through the diffusing pipe F to a sealing tank. Under normal conditions and with the pump supplying the water running at the same speed as the circulating water pump, the quantity of water required is equal to about 10 per cent of the circulating water.

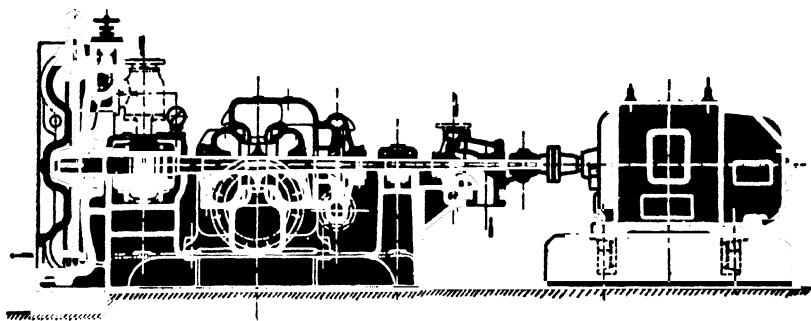


FIG. 182.

COMBINED PUMPING UNIT FOR CONDENSING PLANT.

(Brown-Boveri, A. G.)

As in other hydraulic ejectors with single jets, the diameter of the jet is so large that it is not likely to choke up, as any obstacles which will pass the pump will pass the jets.

In Fig. 182 there is shown a section through a pumping set used in connection with a Brown-Boveri hydraulic ejector. Ignoring for the time being the driving turbine and motor at either end, it will be seen to consist of three pumps—

- (a) Main circulating water pump,
- (b) Operating water pump, and
- (c) Condensate extraction pump.

The supply to the operating water pump is taken from the circulating water pump discharge by the internal connection shown. In the pumping set shown, the turbine is intended to act as a stand-by, the motor usually driving the set. The turbine is fitted with a hydraulic pressure control device which is shown in Fig. 183.

The special feature consists of a double-beat valve on a free spindle, upon which is mounted a diaphragm at its lower end. The under side of this diaphragm is connected to the operating water pump discharge. When the plant is standing, the spring shown keeps the double-beat valve open, so that steam, admitted through the hand-operated main stop valve, passes to the turbine. If, however, the stop-valve is closed and the set run up by motor, the water

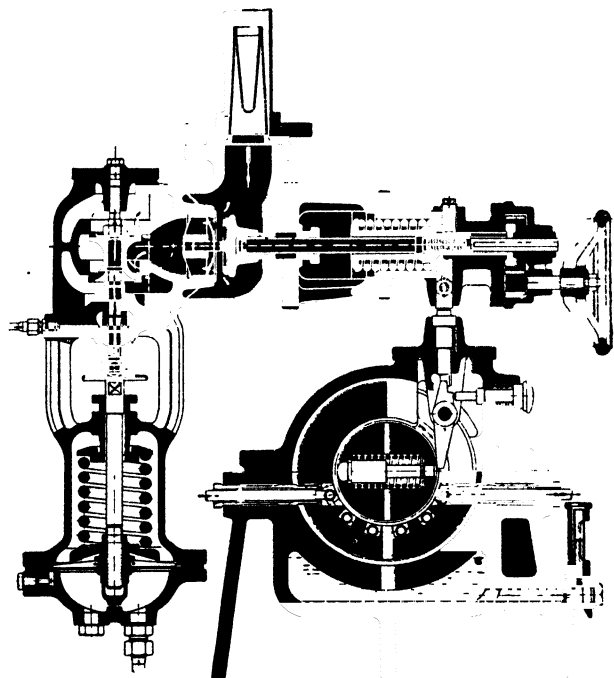


FIG. 183.

BROWN-BOVERI PUMPING UNIT.

(Hydraulic Pressure Control Device.)

pressure under the diaphragm closes the double-beat valve. The stop valve should then be opened sufficiently to pass full-load steam. Should the motor fail, the speed is reduced and the double-beat valve opened. The turbine thus takes up the drive until the motor is back on circuit. This eliminates the stoppages which occasionally arise due to motor failure.

The same device can also be arranged so that part of the balancing pressure can be obtained from a suitable tapping on the main turbine casing instead of all being provided by the spring. When

the load on the main turbine is reduced below a predetermined amount, the double-beat valve closes and a relay brings the motor into circuit. This method may be adopted when the turbine is to be used for the usual drive, the exhaust being used for heating the main condensate. With light loads on the main turbine, there would be a surplus of exhaust steam which is avoided by bringing in the motor to do the driving.

Brown-Boveri have adopted the water ejector to the withdrawal of the condensate from the condenser. As is well known, centrifugal pumps, in order to draw from a high vacuum, must be placed well

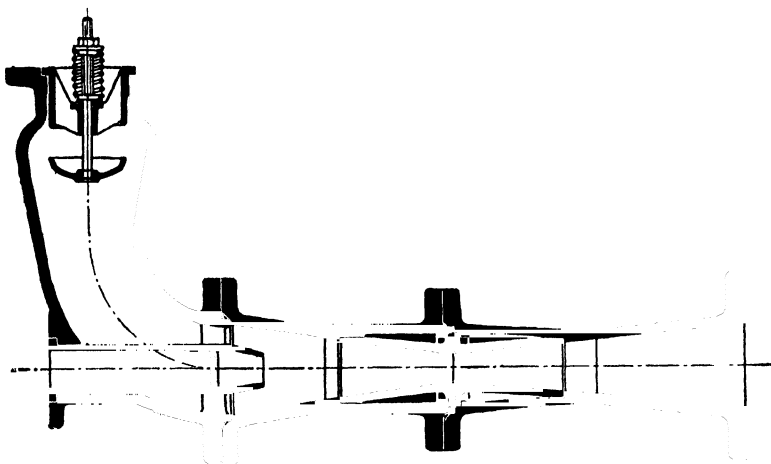


FIG. 184.

HYDRAULIC CONDENSATE EXTRACTOR.

(Brown-Boveri, A. G.)

below the bottom of the condenser. The conditions of the site and the facilities for driving the pump do not always render this desirable. Fig. 184 shows a section through the condensate extraction pipe between condenser and pump. Water taken from the discharge side of the condensate extraction pump passes through the nozzle and combines with the condensate flowing from the condenser. The mixture passes through the usual diffusing tube and tail pipe, and passes to the condensate pump at an increased pressure, this being equivalent to lowering the pump. The arrangement applied to a complete plant is shown in Fig. 185.

TWO-STAGE EXTRACTION PUMP OF THE CONTRAFLO CO. With the same object in view the Contraflo Condenser Co., Ltd., many

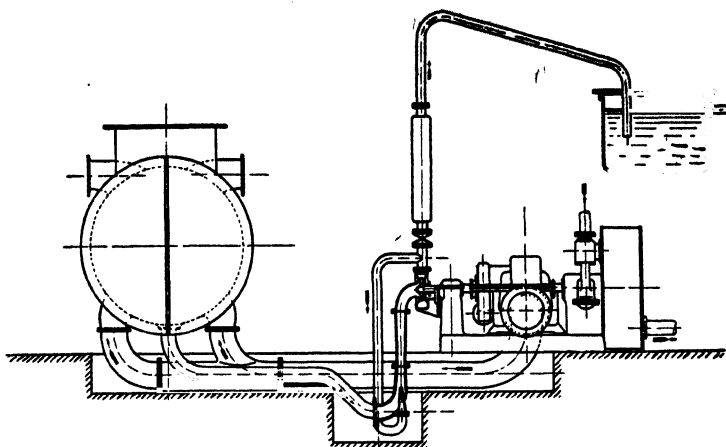


FIG. 185.

ARRANGEMENT OF PLANT WITH HYDRAULIC CONDENSATE EXTRACTOR.

(Brown-Boveri, A. G.)

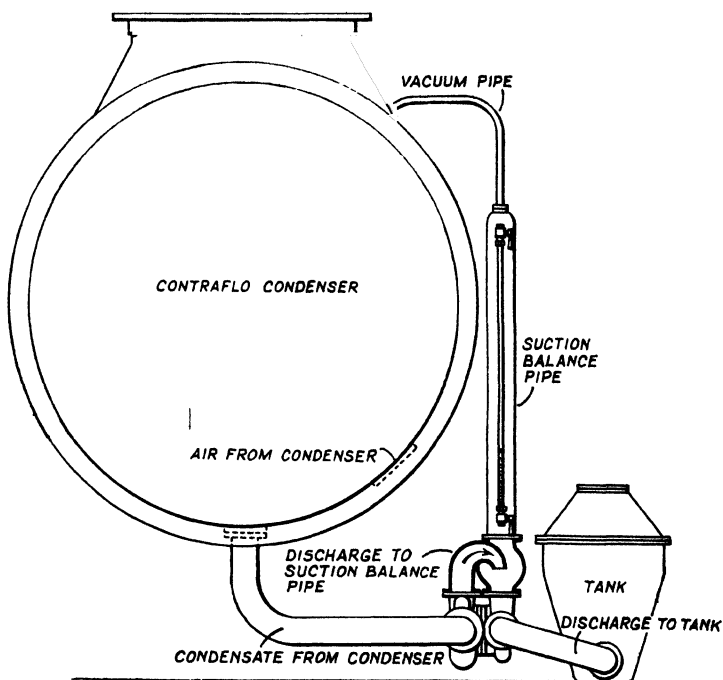


FIG. 186.

TWO-STAGE EXTRACTION PUMP, DIAGRAMMATIC ARRANGEMENT.

(Contraflo Condenser Co.)

years ago introduced a two-stage centrifugal pump with the inter-stage space connected to the condenser at a high point. This is shown diagrammatically in Fig. 186. In this arrangement the first stage is working in the condenser pressure and lifting the condensate to a certain height in the connection to the condenser. This is the

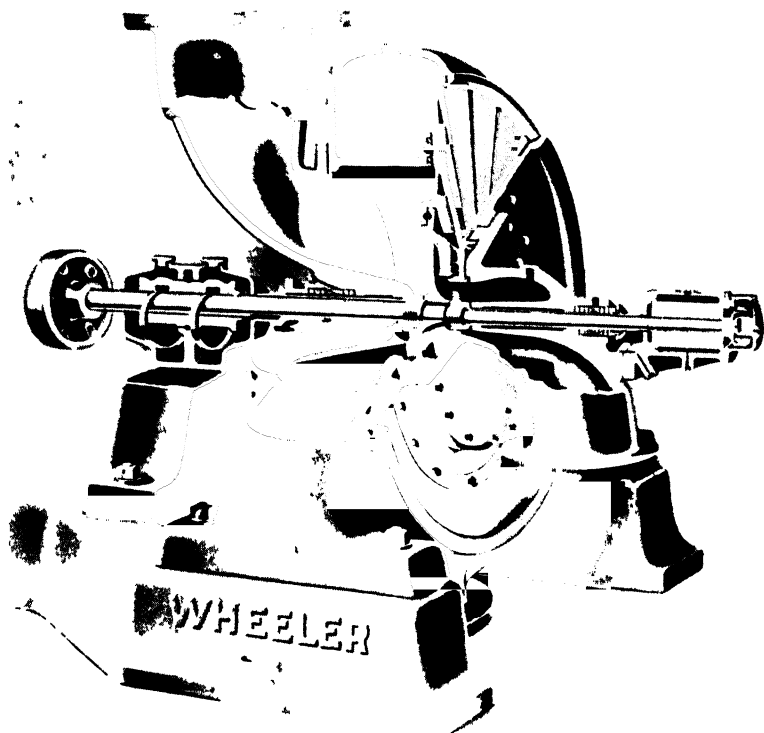


FIG 187

WHEELER ROTARY AIR EXTRACTOR

positive head (over the condenser absolute pressure) on the second stage, and the arrangement is equivalent to sinking the pump by the amount the water is so raised.

The condensate hydraulic ejector previously described performs the function of the first stage of this two-stage pump.

The Wheeler Rotary Air Pump. Still another type of rotary air pump is illustrated in Fig. 187. The method of operation is indicated more clearly in Fig. 188. The centre of the pump consists

of an ordinary centrifugal pump which hurls the operating water across an annular space which is connected to the condenser. The water then passes into a number of tangential compression channels

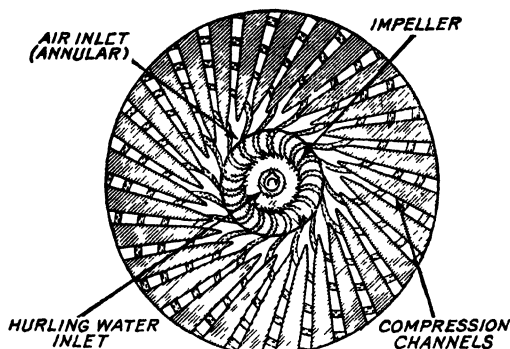


FIG 188
WHEELER ROTARY AIR EXTRACTOR

carrying the air with it. The water passes down each channel as a number of successive pistons in a manner similar to those in the Leblanc rotary pump (see Fig 157).

This pump was introduced by the Allgemeine Elektrizitäts Gesellschaft of Berlin, and has also been made by the Wheeler Condensor and Engineering Co., of Carteret, New Jersey.

CHAPTER XXIII

AIR PUMPS—COMPARISONS OF ROTARY AND EJECTOR PUMPS

EVERY power station engineer when drawing up a specification for new generating plant has to decide upon the type of air extractors most suitable for his particular installation. Whilst it is hardly possible to lay down hard and fast lines of demarcation as to the most useful field for steam or water extractors, it is possible to review the particular advantages of each type. In this comparison the reciprocating pump such as the Edwards will not be referred to in detail, as this type is not used to any extent, particularly with the larger sizes of generating units now installed in so many stations.

A direct comparison of air pumps alone is not possible, as other considerations have to be brought into the account, such as the method of driving the pumps, the method of heating the feed water to render it warm enough to pass into the economizers, the method of introducing the make-up feed water, the maintenance of a closed circuit between condenser and boiler, the deaerating of the condensate, etc. All these items are linked up with air pump types, and have to be considered in arriving at the maximum overall efficiency of the plant, i.e. the minimum cost, including both running costs and capital charges, per unit of electricity generated, per pound of yarn produced, or per unit of product generally.

Methods of Heating the Condensate. The necessity and importance of introducing some method of heating the condensate was emphasized when vacua of over 28 ins. were generally adopted. Many plants are now installed designed to maintain a vacuum of 29 ins. or even higher. The temperature corresponding to this vacuum is 78.9° F., and the condensate will not be at a higher temperature than 77° F. Economizers cannot be fed with water below 100° F., with satisfactory results, so that a small degree of heating is necessary. A much greater degree of heating than that required for the economizer has been adopted widely as by using steam, which would otherwise be condensed by the circulating water, to heat the condensate, heat is preserved which would have been lost to the system.

There are two possible sources of steam for feed heating—

(1) Steam from auxiliary turbines driving the circulating, air (if rotary), and condensate extraction pumps.

(2) Steam abstracted or tapped from the main turbine at a suitable stage in the expansion.

A third source might be mentioned, although it is a variation of the second source. This is a separate auxiliary turbine driving a generator to supply the auxiliaries in the station and exhausting into a special condenser through which condensate from the main turbine passes as circulating water. Such a set is generally called a house service turbine.

If the circulating pump be driven by a steam turbine, the amount of exhaust steam available will usually be considerably more than that required for feed heating, hence, judged solely from this aspect, the circulating pump should usually be driven by a motor.

With a rotary air pump and the condensate extraction pump, both driven by a steam turbine, the steam available in the exhaust can generally be used for feed heating provided that due precautions are taken with regard to light load conditions.

In cases where a rotary pump is not used, the only turbine driven pump is the turbine extraction pump, but the steam used in the steam-jet ejector is then available for feed water heating.

It is quite usual now to heat the condensate up to 250° F. to 300° F. using steam tapped from the main turbine, with or without steam exhausted from the pump turbines. Heating to this temperature, comparable with the temperature of evaporation, has a twofold advantage

- (1) It reduces the heat rejected to the circulating water, and
- (2) It reduces the temperature range on the boiler.

A further effect upon the efficiency of the low pressure end of the turbine will not be considered in detail here as it is not a condensing plant problem, but it should be taken into account in a complete examination of any particular case.

In order to crystallize the various alternatives, an actual case will be considered, but it should be remembered that the assumptions here made may not be applicable to other cases.

Take a 10,000 kw. turbine exhausting into a condenser with vacuum of 28.5 ins. with steam at a pressure of 250 lbs. per sq. in. gauge superheated to 700° F., the consumption being about 103,000 lbs. per hour. The quantity of dry air for which the pumps would be designed would be 54.5, say 55 lbs. per hour. For a fair comparison the two systems must be so arranged that the feed water is heated to the same temperature in both cases.

The rotary pump will, initially, be assumed to be motor driven, and the steam required to heat the condensate will be tapped from the main turbine.

In the ejector about 4 per cent of the heat supplied is actually used to compress and discharge the air, and the remainder used to heat the condensate in the inter- and after-condensers. If the condensate, after heating in the inter-condenser, is returned to the main condenser through a loop pipe, as is frequently done with inter-condensers of the jet type, some of this heat will eventually be carried away by the circulating water. It will be assumed that the inter-condenser is of the surface type, and so the whole of the 96 per cent of the heat supplied to the ejector is retained by the condensate if we neglect radiation and leakage losses.

To equal this heating with the rotary pump, steam must be tapped from the main turbine. Steam could be tapped at many points on the turbine, but it is more economical to allow the greater part of the heat drop to be used in the main turbine and so obtain the greatest possible output. As more work has to be done, the heat remaining per pound of steam will be less than the heat utilized per pound of steam in the ejector, so that more steam is required to be tapped with the rotary plant than the amount supplied to the ejector. In order to retain the comparison on a fair basis the ejector must be credited with the output, which could be obtained in the main turbine with the extra amount of steam.

Dealing with the rotary pump first, the pump will absorb 55 h.p., and it will be assumed that the condensate extraction pump takes the same power in both cases. This extra power absorbed by the rotary pump, 55 h.p., is equivalent to an input of 46 kw. The steam consumption of the ejectors for this duty will be about 1,320 lbs. per hour, and 4 per cent of this will be used for discharging the air, leaving about 1,267 lbs. available for heating the feed.

It will be assumed that in both schemes, the condensate has already been raised in temperature to 150° in a separate heater using either steam tapped from the main turbine or from some auxiliary exhaust.

The heat available per pound is 1,160 B.Th.U., and this is sufficient to raise the whole condensate about 14.2° F. In the rotary scheme sufficient steam has to be extracted from the turbine to give the same heating effect. Owing to the steam having expanded and done work in the turbine, the heat available per pound of steam is less than 1,160, say 990 units. With this heat contents, 1,535 lbs. of steam will be required to raise the condensate 14.2° F., or 215 lbs. per hour more than the steam used in the ejectors. Down to the point of abstraction, this 1,535 lbs. of steam will have generated 112 kw. The 215 lbs. with which the ejector is to be credited, will generate during its complete expansion in

the turbine, 21 kw. Hence, neglecting the heating which is the same in both schemes, in the rotary scheme 112 kw. are produced and 46 kw. used, or a net output of 66 kw., and in the ejector scheme the net output is 21 kw., so that with steam supply and feed heating constant, the output in the rotary pump scheme is greater by 45 kw. This is equivalent to 0.45 per cent of the output of the main turbine.

This shows that with equal steam supply and equal condensate heating, the net output for the rotary pump is approximately three times that of the ejector scheme.

Similar calculations for vacuum of 27.5 ins. and 29.2 ins. show gains of 0.23 per cent and 1.42 per cent. in favour of the rotary pump.

Advantages of Each Type of Pump. These calculations show that the rotary pump has an advantage as regards economy in comparison with the ejectors. Steam ejectors have other compensating advantages, sufficient frequently to justify their selection. Thus there are no moving parts at all beyond the condensate extraction pump which is common to both systems. The ejectors can be attached to the side of the condenser as shown in Fig. 130, thus leaving the floor space clearer. The only portion subject to wear is the steam nozzle which can be renewed easily. They are simple to start and to operate. The first step after draining the apparatus is to open the main steam valve, which is generally arranged to admit steam directly to the second stage. When the vacuum is about 15 ins., water is admitted to the inter-condenser (if not of the simple type) and steam admitted to the first stage. In closing down this sequence of operation is reversed. A further advantage is that the steam ejector does not make any demand upon the output of the main generating plant as is done by a motor-driven rotary pump. A steam-driven rotary has not this disadvantage, as, like the ejector, it makes its demands directly on the boiler. Hence, with all these advantages and disadvantages on both sides, the final selection of either rotary pump or steam ejector is largely a matter of the personal opinion of the responsible engineer. In many cases when rotary pumps are adopted, a steam-jet ejector is also fitted as a reserve, and also to facilitate rapid starting of the plant as shown in Fig. 172.

CHAPTER XXIV

AIR PUMP DRIVES

Motor or Turbine Drive for Auxiliary Pumps. In order to compare the economy of driving the rotary pump by motor or turbine, assume that the rotary pump considered on page 302 is now being driven by a small turbine, exhausting against atmospheric pressure. The efficiency of this turbine will not be more than 40 per cent, so that the consumption will be about 24 lbs. per horse-power per hour, and the contents per pound at discharge about 1,177 B.Th.U. measured from water at 150° F. The total consumption of the turbine will be 55×24 , or 1,320 lbs. per hour, and the exhaust steam will raise the condensate of the main turbine (103,000 lbs. per hour) by

$$\frac{1,320 \times 1,177}{103,000} \text{ or } 15.1^\circ \text{ F.}$$

To compare the motor driven pump equitably, steam must be tapped from the main turbine to raise the main condensate by this amount. Expanding the steam as was assumed previously, the heat units at exhaust will be 990, and the steam to be tapped will be

$$\frac{103,000 \times 15.1}{990} = 1,570 \text{ lbs. per hour,}$$

This amount of steam will generate down to tapping point about 114 kw. But only 1,320 lbs. were supplied to the turbo-driven pump, so that to be on equal terms the difference of 250 lbs. which will generate 24 kw. must be credited to the auxiliary turbine.

Hence the final comparison is as follows: 1,570 lbs. of steam per hour are supplied in connection with both schemes. In the turbo-driven pump 1,320 lbs. are used in the auxiliary turbine, and when exhausted raise the condensate temperature by 15.1° and 250 lbs. in the main turbine generate 24 kw. In the motor-driven pump, the whole 1,570 lbs. are used in the main turbine where they generate 114 kw. prior to exhausting. After being exhausted this steam raises the temperature of the condensate by 15.1°. The motor absorbs 46 kw. to drive the pump, so that the turbo-drive produces 24 kw. and the motor drive 114 - 46 kw. or 68 kw., a net gain of 44 kw., the heating being the same in both cases.

At one time another method was used to compare the alternative systems of driving the auxiliary pumps. It was then stated that

a rise in the condensate temperature of 15° would reduce the coal consumption by

$$\frac{15.1 \times 100}{1,377} = 1.1 \text{ per cent}$$

The steam used in the auxiliary turbine is equal to

$$\frac{1,320 \times 100}{103,000} = 1.28 \text{ per cent}$$

so that the net consumption is 0.18 per cent, this corresponding on a 10,000 kw. turbine to 18 kw. The motor drive required an input of 46 kw., so that this method shows the turbine as the superior by 28 kw. contrary to the previous method.

The first method appears to be the most rational as it stipulates equal thermal conditions, i.e. the same supply of steam and the same temperature of discharged condensate.

This apparently decided advantage of the motor-driven pump over both the turbine-driven pump, as shown by the former method of comparison, and the steam ejector, is presumably due to the fact that the steam supplied to the main turbine is used in a highly efficient turbine, whereas the auxiliary turbine cannot be so efficient. The efficiency of the latter was assumed to be 40 per cent (measured on the ideal Rankine cycle) and that of the main turbine may be as high as 75 per cent. Allowing for generator efficiency of 95 per cent, transmission loss of, say, 2 per cent, and a motor efficiency of 89 per cent, the overall efficiency of the motor-driven pump is about 63 per cent. Thus for every kilowatt generated in the auxiliary turbine the same weight of steam used will generate and deliver to the motor an input corresponding to an output of 1.575, and this excess of 0.575 kw. is the advantage which the motor drive holds over the turbo drive.

Considerations of Capital Cost. This method of comparison leaves entirely out of account the capital cost and charges of the two methods. The turbine is more expensive than the corresponding motor, but whereas the turbo drive only makes a demand upon the boiler capacity, the motor drive requires a certain proportion of the capacity of the main turbine and generator, and switchboard, etc., as well as the boiler. A proper allocation of the charges on these various components of the plant would reduce the advantage held by the motor drive due to its higher efficiency as shown above, and its lower capital cost as compared to the auxiliary turbine.

Considerations of Starting Conditions. Another point which has to be considered is the starting procedure. If the plant is isolated

and the plant shut down, it is more convenient to have steam-driven auxiliaries, so that the condensing plant can be started up and a vacuum obtained before running the main turbine. These conditions may obtain in a small industrial plant, but are very rare in a large central power station. Even in small isolated stations, a motor drive can be used if continuous current be installed and if storage batteries are provided. The larger stations are now so frequently linked up together that current is generally available for auxiliary motors at all times. The larger the sets that are installed in the stations, the more important it is that the turbines are not run non-condensing, and to ensure that the need of doing so does not arise. Thus it has become to be a recognized practice to install for each two or three main generating sets of 10,000 kw. or upwards, a 600 to 1,000 kw. house service turbine. This smaller turbine exhausts into a feed heater at, or slightly below, atmospheric pressure, and this can be started in a standing station. Until there is condensate from the main turbines to pass through this heater, the service turbine can exhaust to atmosphere and thus current is available for running motor-driven auxiliaries.

Although each case requires special consideration, it is probably safe to say that motor-driven auxiliaries will be the most economical under the average conditions prevailing in a modern central power station. For isolated plants such as the power plant in small works and factories steam-driven auxiliaries may prove better than continuous current and batteries.

Heating Condensate by Steam Tapped from the Main Turbine. One factor which has had an important bearing upon the method of driving is the greater use now made of steam tapped from the main turbine for feed heating. The subject is more closely allied to the turbine than to the condensing plant, but reference must be made to it. This method of tapping steam enables some of the latent heat, otherwise rejected to the circulating water, to be diverted to the condensate and so to be returned to the boiler. On the assumption that each pound of steam tapped contains 1,000 B.Th.U., then each 1 per cent of steam tapped will raise the condensate temperature by 10° F. If it be desired to raise the condensate temperature by 150° F. which is quite usual, the steam required is a little below 15 per cent. Assuming that the condensate temperature was 90° F., then the desired final temperature is 240° F. To have a reasonable temperature difference the tapped steam should have a temperature of at least 250° F., that is, it should be at a pressure of approximately 15 lbs. per sq. in. gauge and preferably without superheat. To abstract at this pressure sufficient steam to do the whole of the

heating would not be economical. If it were done the mean temperature difference would be very great, as the temperature difference would be 160° at one end and 10° at the other, as shown in Fig. 189.

AB represents the total steam abstracted for feed heating, and the curve CD shows the condensate temperature rise through the

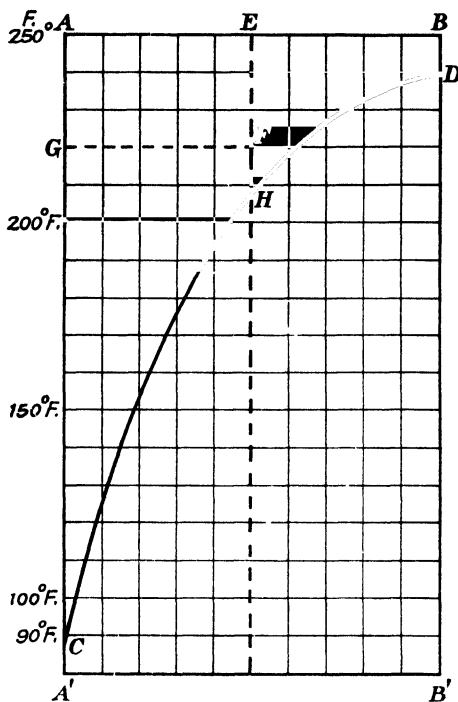


FIG. 189
TEMPERATURE DIAGRAM

feed heater. From A to E , the temperature difference is greater than it need be, and for this portion of the heating, steam could be abstracted at a lower temperature G , and still leave a workable temperature difference. Thus a portion of the steam could be abstracted at, say, 220°F . instead of 250° , and so be enabled to do more work in the main turbine.

Heating Condensate by Exhaust Steam. If there is any exhaust steam from the auxiliaries it should be used to do the last portion of the heating. Assuming that the exhaust steam is sufficient in quantity to raise the condensate by an amount equal to the difference

between H and D in Fig. 189, then it is obviously more economical to confine the steam tapping to the lower temperature and save that steam represented by EB , than to reduce, in any way, the tapped steam represented by GF .

This auxiliary steam may come from small auxiliaries which can conveniently be designed to discharge at high back pressure, or it may come from a house service turbine as stated previously. A house service turbine if of sufficient size could be made to exhaust

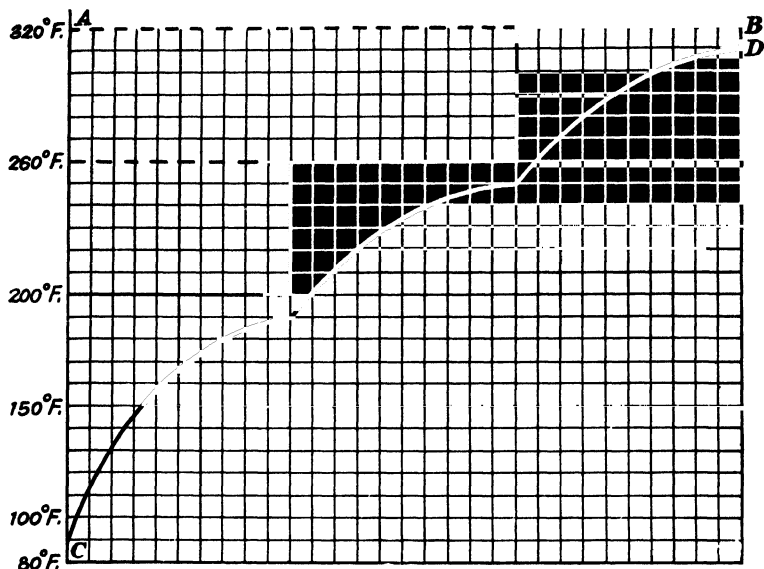


FIG. 190.

IMPROVED TEMPERATURE DIAGRAM.

at two pressures, or steam could be tapped off at a higher pressure than the final exhaust as this would improve the efficiency of operation. With a house service turbine thus arranged, it should be possible to have enough steam to give the desired temperature, and so avoid the necessity of having tapplings on the main turbine at all.* This however, only applies to the larger sets, and for medium plants tapping has resulted in considerable economies, and has rendered the motor drive the more desirable.

Reverting to Fig. 189, it should be noted that by abstracting steam for the portion from A to E at a lower pressure, the heat

* In the new Rimmelsburg Station, Berlin, the main 70,000 kw. sets are not tapped, all tapping being done on 10,000 kw. house service sets which also deliver to the main busbars.—*Z.V.D I.*, October, 1925.

contents per pound of steam are reduced, hence a greater weight of steam is required and, further, as the mean temperature difference is also reduced, a greater surface must be provided in the heater.

In many cases it is desired to raise the temperature of the condensate to, say, 300° to 310° F., and to effect this a third tapping must be provided at a temperature of about 320° F. or about 75 lbs. per sq. in. gauge. The temperature diagram then becomes as shown in Fig. 190.

The effect of tapping steam upon the steam consumption of the turbine will not be discussed as it is not a condensing plant problem.

Heat Balance. This subject naturally leads up to the complete heat balance of turbo-generating plants, a subject too great to consider here. Several papers have been published showing the methods adopted in some large power stations for attaining the most economical heat balance.

CHAPTER XXV

TESTS ON CONDENSING PLANT AND AUXILIARIES

IN this chapter there have been collected details of various tests both on complete plants and on air extractors, together with certain deductions therefrom.

Müller Hydraulic Air Extractor. In the following table there are given some data relating to air pumps, by Daniel Adamson & Co., Ltd., working on the Müller principle, for details of which see pages 281 to 289.

No.	Nozzle Outlet Diameter.	Diffuser Throat Diameter.	Water Press in Ft.	Quantity in G.P.M.	Vacuum.	Water Temp. ° F.	Lbs. of Dry Air per Hour.
1	3"	4 ¹ / ₄ "	28	1,000	28"	80	18
2	3 ³ / ₈ "	4 ⁷ / ₈ "	30	1,300	28"	80	28
3	3 ³ / ₈ "	5 ¹ / ₈ "	43	1,400	28"	80	35
4	3 ¹ / ₂ "	4 ¹ / ₂ "	30	1,150	27 ¹ / ₈ "	80	36
5	2 ¹ / ₈ "	2 ¹ / ₈ "	28	500	27 ¹ / ₂ "	80	14

Examining the nozzles first, we can calculate the velocity of the water issuing from the jets by the formula—

$$\text{Velocity} = C \sqrt{2gh}$$

when h is the total effective head, and the quantity passed by the nozzle is equal to Area $\times C \sqrt{2gh}$.

Applying this to the above five cases, the value of C or the nozzle velocity coefficient is as follows—

No	1	.	.	.	.	0.876	} Average value = 0.877
	2	.	.	.	.	0.886	
	3	.	.	.	.	0.867	
	4	.	.	.	.	0.882	
	5	.	.	.	.	0.887	

In calculating the average velocity coefficient no allowance has been made for the stationary scroll placed in the nozzle to give the water a twist. By imparting a spin to the water without adding any energy to it, the axial velocity of the water is reduced, thus apparently reducing the velocity coefficient.

The nozzle energy coefficient is equal to the square of the velocity

coefficient, the energy being proportional to the square of the velocity. The energy supplied in case No. 1 is equal to

$$\frac{1000 \times 10 \times \left(28 + \frac{28 \times 13.6}{12}\right)}{60} \text{ ft.-lbs. or } 9,950 \text{ ft.-lbs. per sec.}$$

The loss of energy is

$$9,950 (1 - 0.876^2) \text{ or } 2,310 \text{ ft.-lbs.}$$

The actual energy supplied from external sources to the jet is that corresponding to the actual water pressure, or in this case

$$\frac{1000 \times 10 \times 28}{60} \text{ or } 4,666 \text{ ft.-lbs. per second.}$$

The difference between 9,950 and 4,666 ft.-lbs. is supplied by the flow into the vacuum space, but as an equal amount of energy is absorbed in discharging the water back to the atmosphere this can be omitted from the calculations. Hence, 4,666 ft.-lbs. are supplied, of which 2,310 ft.-lbs. are lost in the nozzle, leaving 2,356 ft.-lbs. to do the actual work of compressing and discharging the air. Dealing similarly with the other four examples we have the following—

No	Energy Used to Deal with Air in Ft.-lbs. per Second	Capacity in Lbs. of Dry Air per Hour	Vacuum
1	2,356	18	28
2	3,612	28	28
3	5,703	35	28
4	3,140	36	27.5
5	1,200	14	27.5

To reduce the two latter to a comparable basis, they must be corrected for vacuum. As the determining feature is the volume to be handled, the energy is to be increased in the ratio of the specific volume of air at the two vacua, or in the ratio of $\frac{421.4}{277.7}$.

With this increase, the energy in Cases 4 and 5 becomes 4,764 and 1,824 ft.-lbs. respectively. The relation between energy per second and air capacity, is shown in Fig. 200, and it will be seen that Cases 1, 2, 4 and 5 lie practically on a straight line, but that Case 3 is considerably above the line. The relation is such that approximately 133.5 ft.-lbs. per second have to be liberated at the exit from the nozzle for each pound of air handled per hour. The water pressure used in Case No. 3 is much greater than in the others,

and it appears probable that this plant had a considerable margin. Possibly it was designed for 25 per cent overload conditions.

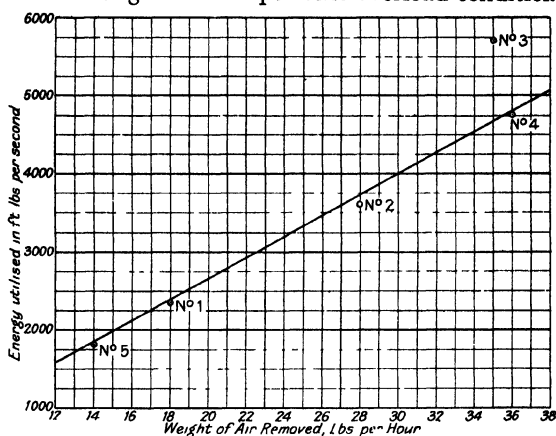


FIG. 191.

MULLER EJECTOR TEST

(D Adamson & Co., Ltd)

In Fig. 167 there are shown the details of a Muller ejector made by the Grevenbroich Co. which was supplied with 2,028 gallons

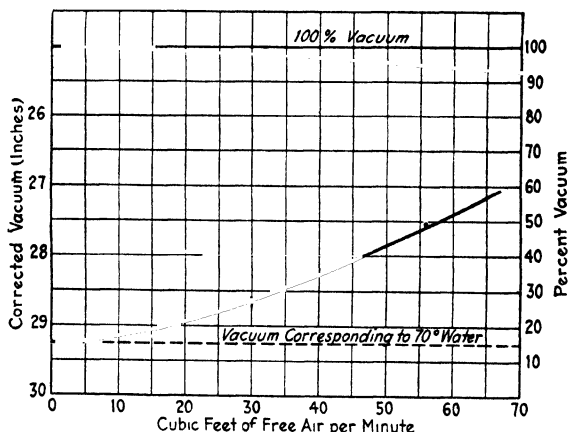


FIG. 192

per minute at a pressure of 34.4 ft. The vacuum maintained was 27.6 ins. With a nozzle diameter of 104 mm. or 4.1 ins. the theoretical flow under these conditions is 2,220 gallons per minute, so that the nozzle velocity coefficient is, in this case, 0.912, or slightly higher

than that found for the Adamson ejectors mentioned previously. This variation may be due to the angle of the scroll vanes, to the method and place of measurement of the water head, etc.

The design of this type of plant would be facilitated by the preparation of reference tables, giving the foot-lbs. of energy per second at the nozzle exit per 100 gallons of water per minute when discharged from various heads into various vacua. The following is shown as an example, being based on 28 and 30 foot water pressures, the temperature being 80° F.

Vacuum.	Water Pressure.	
	28'	30'
27·0"	238·8	264·5
27·5"	236·6	262·3
28·0"	234·5	260·1
28·5"	232·2	257·9
29·0"	230·0	255·7

Thus to remove 18 lbs. of air per hour, and to maintain a vacuum of 28 ins., the water required at a pressure of 28 ft. would be

$$\frac{18 \times 133.5}{234.5} \times 100 \text{ gals. per min.}$$

or 1,025 gallons per minute. If water at 30 ft. head be used, the quantity is

$$\frac{18 \times 133.5}{260.1} \times 100, \text{ or } 924 \text{ gals. per min.}$$

In determining the actual pressure to be used, and the corresponding quantity, there are several factors to be taken into consideration. A high head with a reduced quantity, whilst fulfilling the requirements as regards energy supplied, might not present a sufficiently stable entraining surface. Again, the makers have standard sizes of nozzles, and by adjusting the pressure a standard may be used. Another factor is the source of the operating water. If it be taken from the discharge side of the circulating pump in a plant working with cooling towers, the total head on the pump will be in excess of 28 ft. to 30 ft. and so will be ample. If a separate pump be employed to supply the operating water, the head can be selected to suit a standard nozzle. As in most engineering problems, the selection of operating water head and quantity is a matter of compromise.

In a Müller, as in other hydraulic ejectors, the maximum theoretical vacuum is that corresponding to the temperature of the operating water. In Figs. 192 and 193 there is shown the variation of absolute pressure with increase in air inflow, this absolute pressure being in excess of that corresponding to the temperature of the operating water.

From Fig. 193 it will be seen that this particular ejector which was supplied with 1,400 gals. per minute at a head of 48 ft., will maintain an absolute air pressure of one inch of mercury when dealing with 35.6 lbs. of air. If the water temperature is 80° the

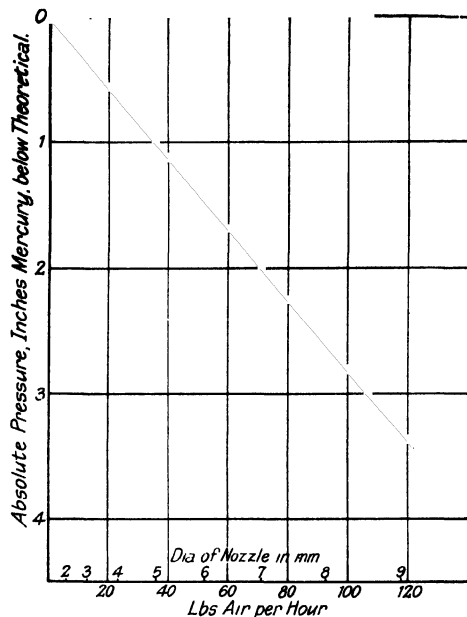


FIG 193

VARIATION OF ABSOLUTE PRESSURE WITH INCREASE IN AIR INFLOW.

vapour pressure is also one inch and the vacuum therefore 28 ins. Similarly, if the water temperature be 60°, the vacuum would be 28.5 ins. In the following table there is another method of showing the effect of the temperature of the operating water

TABLE XIII

Vacuum	Air Capacity in Lbs per Hour	
	Temperature of Operating Water	
	60°	80°
29"	18.1	Nil
28"	55.7	35.6
27.5"	88.7	71.9
26.5"	106.9	90.6
26"		106.9

Assuming that the nozzle velocity coefficient is 0.877, as determined earlier, the tests shown in Fig. 193 give the following values for the foot-lbs. at the nozzle exit per pound of air handled.

Vacuum.	Foot-lbs. at Nozzle Exit per Lb. of Air.	
	Temperature of Operating Water.	
	60°	80°
29"	378.1	
28"	128.2	194.0
27"	78.5	96.9
26.5"	65.4	
26"		65.7

These values are in excess of those found earlier and indicated in Fig. 191, when an average for 80° water and 28 ins. was 133.5 ft.-lbs. With the exception of test No. 3, the water pressure in that series of tests was 28 ft. to 33 ft. No. 3 with a water pressure of 43 ft. gave a consumption of 163 ft.-lbs., and the above table for 28 ins. and 80°, 194.0 ft.-lbs., with a head of 48 ft. These figures indicate that the efficiency increases as the head is reduced, which is to be expected. Imagine that in two ejectors the energy released at the nozzle end is the same, but in one the quantity of water is double that in the other. In this one the scrolls of water will be thicker and more stable with a better extracting capacity.

HYDRAULIC EJECTOR MORE EFFICIENT THAN STEAM EJECTOR. Considering this hydraulic ejector from its efficiency standpoint, it is seen to be very poor, but this is not a matter of vital importance. The actual work done in compressing one pound of air isothermally from a vacuum of 28 ins. (Bar 30 ins.) to atmospheric pressure of, say, 15 lbs. per sq. in., according to formula given on page 206, is 162,400 ft.-lbs. In Case No. 1, the energy supplied is that corresponding to 1,000 gals. per minute at a head of 28 ft., so that the efficiency is

$$\frac{18 \times 162,400 \times 100}{1000 \times 10 \times 28 \times 60} = 17.4 \text{ per cent.}$$

Although low when compared to mechanical efficiencies of machines, this efficiency is greatly superior to that of the steam jet ejector, which as shown on page 207 has an efficiency of about 4.0 per cent. This is partially explained by the fact that the flow of the water under a head increased due to the vacuum is a reversible process, as there is no physical change in the operating medium—

water, whilst in the steam jet, the process is not reversible as the heat contents of the steam has been degraded to a lower thermal level.

The cycle in this hydraulic ejector is shown diagrammatically in Fig. 194 for Case No. 1.

Time to Establish Vacuum. One point of considerable importance, and one regarding which the early hydraulic pumps did not show to advantage, is the time taken to establish the working vacuum. At the outset it should be noted that to enable a fair comparison to be made between two plants of even the same nominal size,

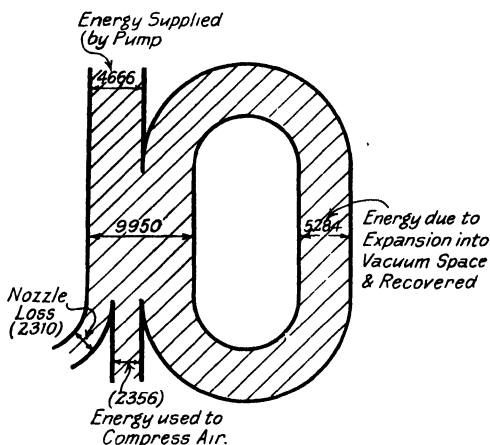


FIG. 194.

CYCLE IN HYDRAULIC EJECTOR.

more information is required than a mere statement of the number of minutes required to attain a certain vacuum. When starting from all standing the turbine and the condenser are full of air, and this space has to be evacuated. The turbine may be a small high-speed geared, or a large low-speed direct coupled set with very different volumetric capacities, although of the same nominal output. Again, the condenser, although designed for the same vacuum, may have in one case to use warmer water than in the other, so that its volume is greater. But the air extractors fitted, being proportioned on the steam exhausted by the turbines, will, within narrow limits, be the same in each case as the difference in steam consumption and the normal air quantity to be handled will only be small. Obviously the extractor, working on the low speed turbine and large condenser, will be slower in building up the vacuum than the other extractor.

Another point which may cause plants to vary in the time required to establish the working vacuum is the method of packing the turbine glands. With steam packed glands, they can be made airtight with the turbine standing, but with water sealed glands the seal is not made until the turbine has attained a certain speed.

With hydraulic pumps, a steam ejector is sometimes fitted for starting-up purposes, and to act as a stand-by extractor.

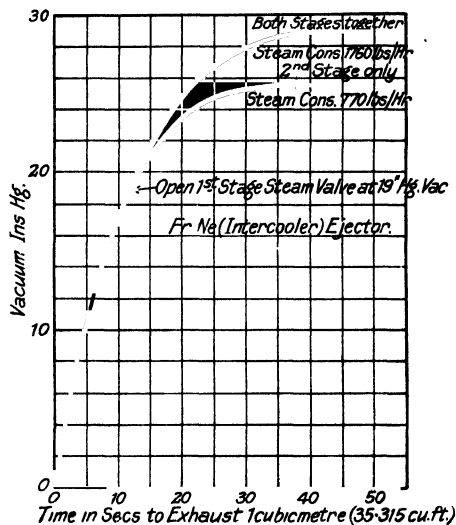


FIG. 195.

M.V. CO. LEBLANC EJECTORS. TIME
TAKEN TO EXHAUST 1 C.U.M.

Hence, the fairest basis for comparison is to note the time taken to build up the vacuum in a stated volume. In Fig. 195 there is shown the rise of vacuum with time, measured in seconds, in a volume of 1 cubic metre, or 35.315 cubic feet. The second stage of the ejector, using 770 lbs. of steam per hour, raises the vacuum to 19 ins. in 12 seconds when the first stage valve is opened. The consumption is then 1,760 lbs. per hour, and a vacuum of 28 ins. was attained in 30 seconds and 29 ins. in 42 seconds.

Fig. 196 shows the rise of vacuum in a 5,000 kw. turbine and a condenser with three types of pumps each of the same capacity—

- (1) Leblanc Rotary with closed pressure tank.
- (2) " " " open elevated tank.
- (3) Steam ejector, simple type, i.e. no inter-condenser.

The first type reaches 28 ins. in just over 10 minutes, and the steam ejector in about 11 minutes. The Leblanc rotary with open tank takes 16 minutes to reach the same vacuum, this showing the advantage of the closed pressure tank for starting purposes when the conditions are similar to those met with in handling heavy air inflows. The same difference is noted in Fig. 197 which refers to a 6,000 kw. set with a 12,500 square foot condenser.

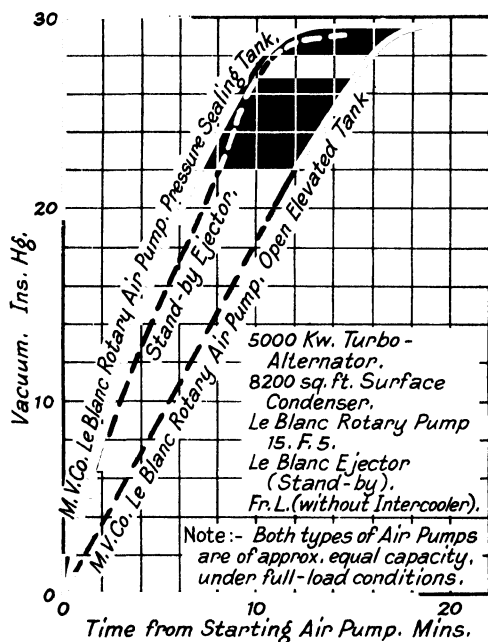


FIG. 196.

TEST ON VACUUM PUMPS OF 5,000 KW. TURBINE.

Tests on Steam Ejectors —EFFECT OF VARYING STEAM PRESSURE

In dealing with tests on steam ejectors it is a difficult problem to get all tests on a comparable basis as there are so many variables which are controlled differently in the different tests.

Before dealing with tests on complete plants, the effect of certain changes on the steam ejector alone will be considered. One of the first problems is to show how the vacuum at the ejector suction varies with the steam pressure at the jet. In Fig. 198 there is shown the result of a series of tests on an ejector designed to use 1,200 lbs. of steam per hour at 150 lbs. per square inch gauge. With most steam jet ejectors it is a simple matter to reduce the

steam pressure at the first stage and leave the second stage unchanged. This has been done in these tests, the first stage pressure ranging from 30 lbs. to 150 lbs., but the second stage pressure was constant at the higher figure. The pressures in the inter-cooler, which was supplied with water at 65° F., are shown at the left-hand end of each curve. It should be noted that the inter-cooler pressure remained constant throughout each test with a constant air quantity, showing that the inter-cooler pressure is a function of the second

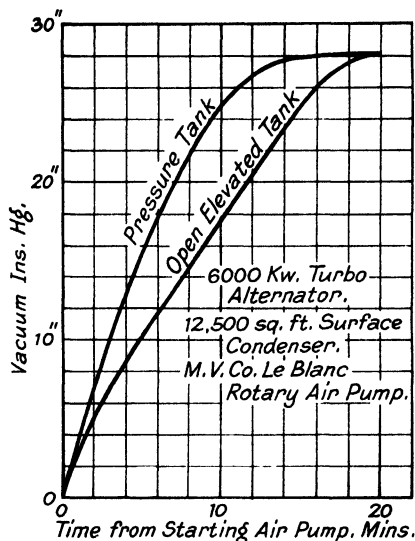


FIG. 197

TEST ON VACUUM PUMP OF 6,000 KW SET.

stage operation and not dependent on the first stage conditions. Thus, with a constant steam consumption in the second stage at a constant steam pressure, the inter-cooler pressures are dependent upon air inflow as follows—

Air. Lbs. per Hour	Inter-cooler Pressure		Plenum on Second Stage.	
	Hg. Mm.	Hg. Inches	Hg. Mm.	Hg. Inches.
23.4	115	4.53	645	25.35
36.5	132	5.20	628	24.68
52.5	147	5.29	613	24.10
71.5	166	6.54	594	23.35
93.5	184	7.25	576	22.64

Now, considering the first stage only, it appears from Fig. 198 that the plenum or pressure increase across the first stage with various pressures and rates of air inflow are as follows—

	First Stage Plenum in Mm. Hg.				
	Air Inflow Lbs. per Hour.				
Steam Pressure on First Stage.	23.4.	36.5.	52.5	71.5.	93.5.
40	90	87	76	92	
60	95	106	113	112	110
80	93	107	117	121	117
100	91.5	105.5	117	120	120.5
120	90	104	116	118.5	122
140	88	103	115	117.25	122
150	87	102	114	116	122
Plenum in second stage	645	628	613	594	576

The actual vacuum, measured in millimetres of mercury with any pressure of steam at the first stage, is the sum of the last line and the figure in the same column corresponding to the selected steam pressure.

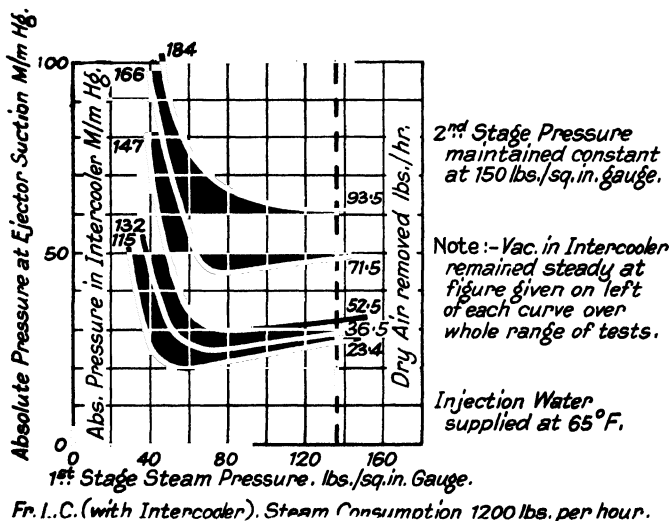


FIG. 198.

PERFORMANCE OF M.V.E. LEBLANC EJECTOR VARYING STEAM PRESSURE.

The first stage plenum is plotted against steam pressure in Fig. 199, which shows that for a given rate of air inflow the plenum increases rapidly with pressure up to a maximum, and then slowly decreases. The decrease in both directions from the maximum is due to different causes.

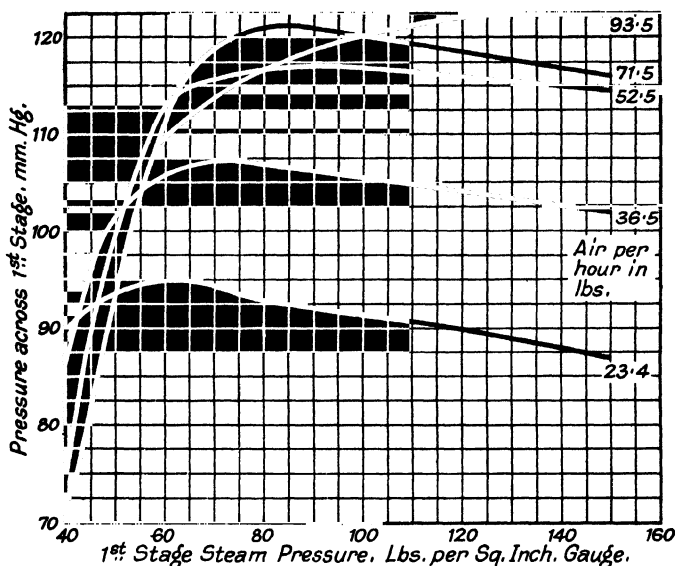


FIG. 199.

PERFORMANCE OF M.V.E. LEBLANC EJECTORS.

The above figures relating to the first stage plenum can be expressed in the form of the ratio of compression in that stage, and Fig. 200 shows this ratio for the plena as detailed above and graphed in Fig. 199.

With steam pressures below that giving the maximum plenum, the velocity of the steam issuing from the jet is insufficient to evacuate the condenser and the absolute pressure rises, until the work to be done is reduced to the capacity of the issuing jet. Increase of steam pressure increases the capacity of the jet with a reduction of absolute pressure. Now with air inflow remaining constant, imagine the steam pressure increased to a value in excess of the value giving the maximum plenum. The steam is moving more rapidly and more steam passes the jet. With a constant weight of air to be removed, the reduction of velocity resulting from the mixing is less than formerly with the result that the

section of the moving central jet of steam and the annular surround of air is reduced. This reduction of area may be sufficient to allow the moving mass to leave the diffuser walls and to allow some air to leak back to the entraining chamber. The higher the steam pressure, the higher is this point of maximum plenum measured

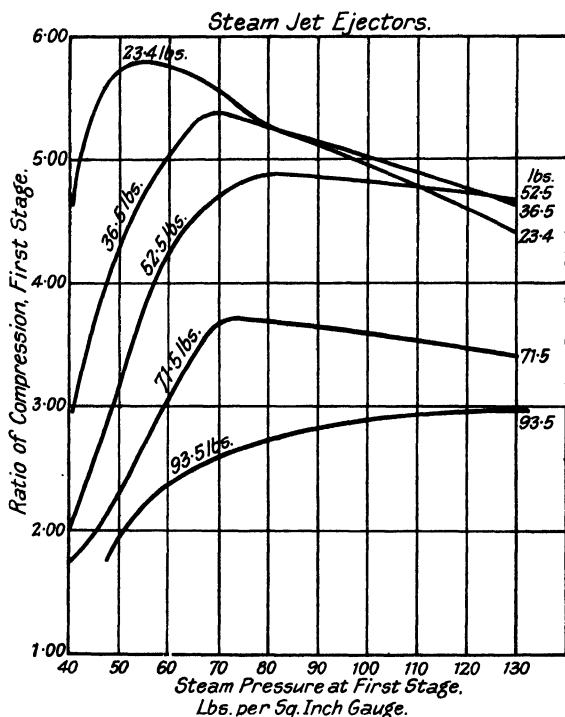


FIG. 200.

PERFORMANCE OF M.V.E. LEBLANC EJECTORS.

in air capacity, as the increased weight of steam will carry an increased weight of air.

These tests show that with any complete two-stage ejector designed to pass a given weight of steam at a chosen pressure, the one fixed point is the pressure against which the second stage has to discharge, this pressure usually being atmospheric or slightly above. The ratio of the steam quantity used in the second stage to the air inflow, determines the plenum created by the second stage, and so fixes the pressure in the inter-cooler, subject to any variation arising from the temperature of the water used in the inter-cooler.

This settles the discharge pressure against which the first stage has to work, and the ratio of the first stage steam to air inflow, as in the second stage, determines the first stage plenum. The sum of these two plena, ignoring the pressure drop, if any, in the inter-condenser, is the vacuum that the ejector will maintain.

It will be noted that the vacuum with a lightly loaded (air load)

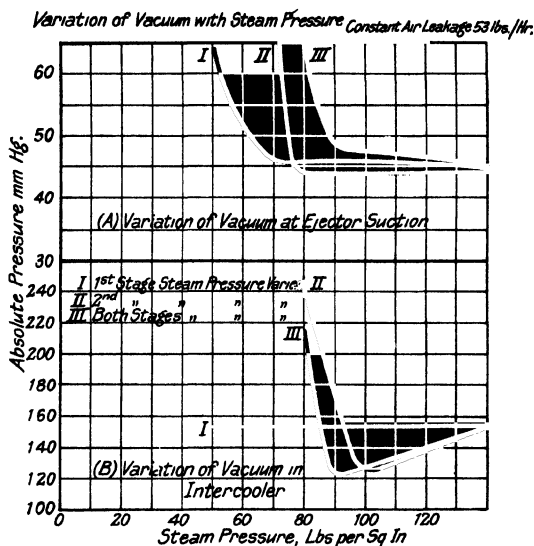


FIG 201

PERFORMANCE OF M V F. I. E. BLANC EJECTORS.

ejector may be improved by reducing the first stage steam pressure. In the test results given on page 346, the first stage pressure is considerably less than that at the second stage.

The steam consumption will be referred to in fuller detail later.

In Fig. 201 are shown the results of some tests in which the steam pressure at both stages was reduced separately and together. The air load is constant at 53 lbs. per hour. The upper curves refer to the vacuum at the ejector suction and the lower ones to the vacuum in the inter-condenser. Lines marked I refer to the case where only the first stage pressure is varied. As before, the inter-condenser pressure is constant. Varying the first stage pressure from 50 to 70 lbs. gauge increases the vacuum from 696 mm. to 714 mm., and further pressure increases have slight effect upon the vacuum.

In curves marked II, the first stage pressure is constant and

the second stage pressure varied, hence the pressure in the inter-condenser varies, the curve of variation being very similar in type to those shown in Fig. 198. From these curves it is evident that the plena and the total vacuum are as shown in Table XIV.

TABLE XIV

Second Stage Steam Pressure. Lbs. per sq. in. g.	Second Stage Plenum. Mm. Hg.	First Stage Steam Pressure. Lbs. per sq. in. g.	First Stage Plenum. Mm. Hg.	Vacuum. Mm.
80	516	140	200	716
90	588	140	128	716
100	634	140	82	716
110	629	140	87	716
120	622	140	94	716
130	615	140	101	716
140	608	140	108	716

In the third pair of curves, the pressure at both stages varies simultaneously, and the plena and vacua are as follows—

TABLE XV

Steam Pressure. Both Stages. Lbs. per sq. in. g.	Plena Mm. of Hg		Vacua.	
	First Stage	Second Stage.	Mm. Hg.	Inches. Hg.
80	151	544	695	27.36
90	75.5	636	711.5	28.02
100	80.8	633	712.8	28.06
110	87	626	713	28.07
120	93.5	620	713.5	28.09
130	101.3	613	714.3	28.13
140	109	606	715	28.15

The rise in vacuum beyond that obtained with 90 lbs. steam is comparatively slight. With a greater rate of air inflow, a higher steam pressure would be necessary to obtain the maximum vacuum.

In Fig. 202* there is shown the increase of vacuum with steam pressure for a Radojet ejector which is illustrated in Figs. 135 to 138. The dotted line shows the figures tabulated above.

Conditions on Starting Up an Ejector. In the above tests the injection water was supplied to the inter-condenser at 65° F. This water is usually condensate and the temperature is considerably higher than this. In such cases a supply of cold water is generally provided for starting-up purposes, this water probably coming

* G. L. E. Kothny. *Society of Naval Architects and Marine Engineers*, New York, November, 1919.

from the circulating water supply. These alternative supplies of water at different temperatures facilitate experiments as to the effect of injection water temperature upon capacity. In an actual

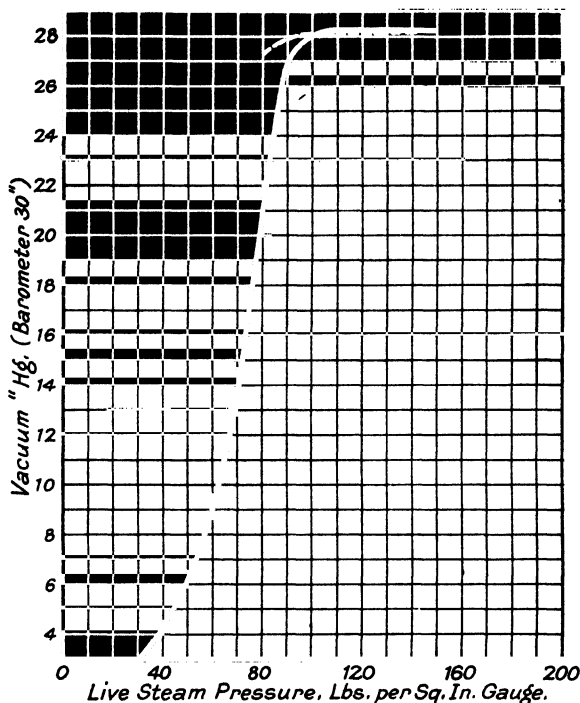


FIG. 202

PERFORMANCE OF RADOJET EJECTOR.

case when the condensate was about 95° F. the capacity was increased by using water at 53° F. as follows—

Absolute Pressure in Inter-condenser Inches. Hg	Increase in Capacity.
1 in	7%
2 in	25%
4 in	28%

This increased capacity is particularly useful when starting up and evacuating the whole condenser and exhaust connections.

The reason for commencing with the second stage when starting-up steam ejectors will be evident from Fig. 203.* In this figure, curve A shows the variation of the condenser vacuum with both

* G. L. E. Kothny. *Society of Naval Architects and Marine Engineers*, New York, November, 1919.

stages working with various rates of air inflow, the steam supply and pressure remaining constant. Curve *B* shows the vacuum maintained when only the second stage is in use. In this case the steam pressure remains constant, but the consumption is less than when both stages are in use. If the steam consumption of the second stage be increased to be equal to that of both stages in curve *A*, the result is shown in curve *C*.

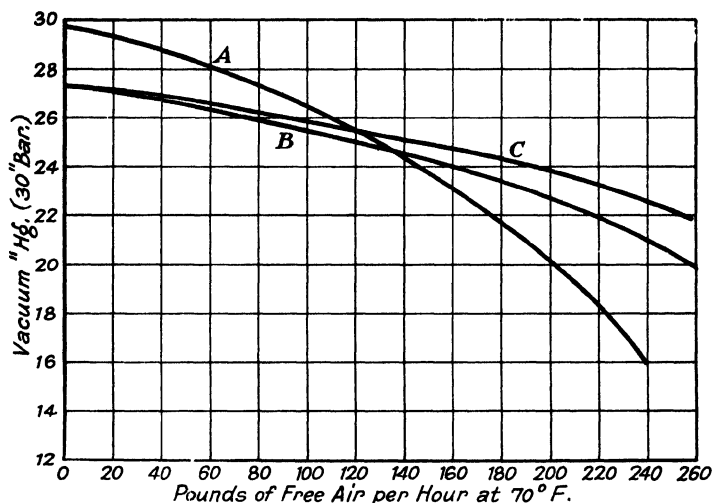


FIG. 203.

PERFORMANCE OF RADOJEI EJECTORS.

These curves show that with both stages working and no air passing the maximum vacuum is 29.75 ins., but with only the second stage the maximum vacuum is only 27.3 ins. With vacua below 24.7 ins. the capacity of the second stage alone is greater than that of both stages as shown by the inter-section of curves *A* and *B*. Hence, when starting up from all standing, the second stage is more effective up to this vacuum. The exact vacuum at which the first stage should be brought into use will vary with different ejectors, but the curves in Fig. 203 are representative.

Similar information regarding a Ljungström two-stage steam ejector has been published* and the curves giving the results are reproduced in Fig. 204. Here it is seen that with absolute condenser pressures in excess of 1.6 ins. Hg. or vacua below 28.4 ins., the second stage alone maintains a better vacuum than do both stages together.

* *Engineering*, 22nd August, 1922.

AXIAL POSITION OF JETS. There is another factor upon which the efficiency and capacity of a steam jet ejector depends, and that is the length measured in an axial direction between the end of the steam jet and the entrance to the diffuser. The greater this distance, the larger the surface presented by the steam jet by contact

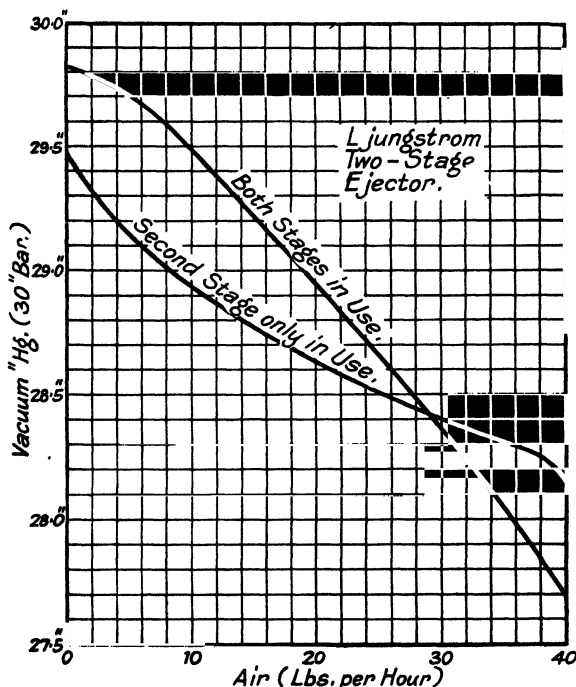


FIG 204

PERFORMANCE OF LJUNGSTROM EJECTOR.

with which the air is removed. This greater length allows the velocity of the outer currents of air to be raised nearer to the velocity of the centre jet, and so equalizes the velocity across the whole entrance to the diffuser. The variation of capacity with this dimension is a point regarding which no reliable test results are available.

Variation of Steam Ejector Capacity with Vacuum. This series of tests upon an ejector with a constant steam supply, Figs. 198 to 201, enable the capacity of an ejector at various vacua to be compared with the theoretical capacity.

If it be assumed that the energy liberated at the nozzle in the

ejector be constant over the complete range, then with constant efficiency the weight handling capacity of the ejector should vary inversely with the work of compression per pound of mixture. The work of compression per pound is proportional to

$$\left(r^{\frac{\gamma}{\gamma-1}} - 1 \right)$$

when r is the ratio of compression, or if the vacuum is v ins., then

$$r = \frac{30}{30 - v}$$

If γ be taken as 1.408, then the work per pound, taking the 28 in. figure as unity, is—

<i>Vacuum.</i>	<i>Relative Work.</i>
29"	1.411
28"	1.000
27"	0.798
26"	0.664

Hence the capacity of an ejector with constant steam supply is the reciprocal of the relative work figures, or—

<i>Vacuum.</i>	<i>Relative Capacity</i>
29"	0.709
28"	1.000
27"	1.235
26"	1.506

If it be assumed that the steam nozzle efficiency is constant with varying vacua (and the variation is insufficient to cause any significant error) then the energy liberated is proportional to the heat drop, and so the capacity is also proportional to the heat drop. Taking the 28 in. figure again as unity, we get —

<i>Vacuum.</i>	<i>Vacuum Correcting Factor</i>
29"	1.100
28"	1.000
27"	0.938
26"	0.892

So that the actual relative capacities corrected for the effect of vacuum upon the jet are—

<i>Vacuum.</i>	<i>Relative Capacities.</i>
29"	0.709 × 1.100 = 0.780
28"	1.000 × 1.000 = 1.000
27"	1.235 × 0.938 = 1.160
26"	1.506 × 0.892 = 1.340

If the figures given on page 334 be graphed and extrapolated down to 26 in. vacuum, we have—

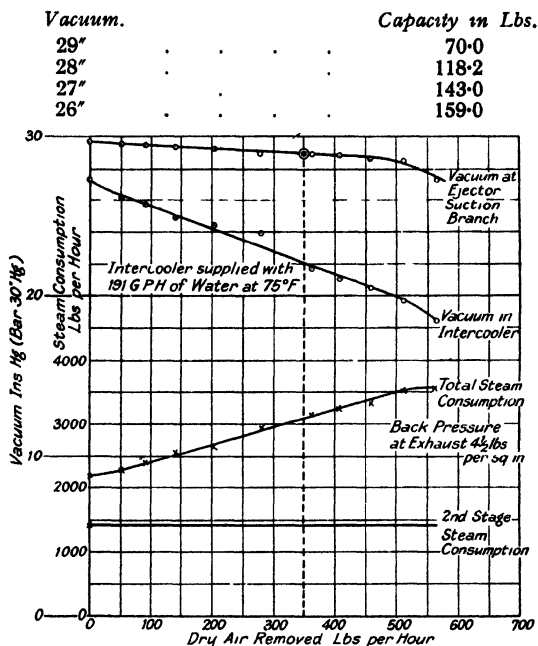


FIG. 205.

STEAM CONSUMPTION OF EJECTOR

Hence the test and theoretical figures are—

Vacuum	Test Capacity	Theoretical Capacity
29"	0.593	0.780
28"	1.000	1.000
27"	1.209	1.160
26"	1.344	1.340

The coincidence of the test and the theoretical figures is reasonably close except at 29 in. vacuum, where apparently the efficiency of the ejector is reduced, an effect which would be anticipated.

Steam Consumption of an Ejector. In all diagrams showing test results given previously, no regard has been paid to the quantity of steam used. In Fig. 205 there is shown the ejector suction vacuum, inter-condenser vacuum, and steam consumptions in both stages for air inflows ranging up to 500 lbs. per hour. The second stage consumption is constant, and consequently the inter-condenser vacuum falls as the air inflow increases. The first stage

steam consumption increases almost proportionally with the air. The figures from the diagram are as follows—

TABLE XVI

Air in Lbs. per Hour.	Ejector Suction. Vacuum. In. Hg.	Inter- condenser. Vacuum. In. Hg.	Consumption : Lbs. per Hour.		
			First Stage.	Second Stage.	Both Stages.
100	29.5	25.6	970	1,430	2,400
200	29.2	24.1	1,270	"	2,700
300	29.0	22.7	1,540	"	2,970
400	28.8	21.4	1,800	"	3,230
500	28.4	19.9	2,070	"	3,500

The above consumption figures are equivalent to the following when calculated in relation to the weight of air handled—

TABLE XVII

Air in Lbs. per Hour.	Ejector Suction Vacuum. In. Hg.	Steam Consumption in Lbs. per Lb. of Air.		
		First Stage.	Second Stage.	Both Stages.
100	29.5	9.70	14.30	24.00
200	29.2	6.35	7.15	13.50
300	29.0	5.13	4.77	9.90
400	28.8	4.50	3.58	8.08
500	28.4	4.14	2.86	7.00

Thus in the above ejector, the total steam consumption varies from 7 to 24 lbs. of steam per pound of air.

The actual ratio of steam used to air removed depends on the size of the ejectors as will be noticed from the following table—

TABLE XVIII

Vacuum at Ejector Suction. Inches.	Lbs. of Steam per Lb. of Dry Air.		
	First Stage.	Second Stage.	Both Stages.
(a) <i>Large Ejector</i> —			
29.5"	12.6	9.4	22.0
29.0"	5.3	3.9	9.2
28.0"	3.62	2.68	6.3
27.0"	3.56	2.64	6.2
(b) <i>Small Ejector</i> —			
29.0"	6.6	7.4	14.0
28.0"	4.23	4.77	9.0
27.0"	3.57	4.03	7.6

These figures are shown graphically in Fig. 206, and enable the consumption for any air load and vacuum to be determined. It must be remembered that the steam supplied to the first stage is controlled by throttling, so that both pressure and quantity vary together.

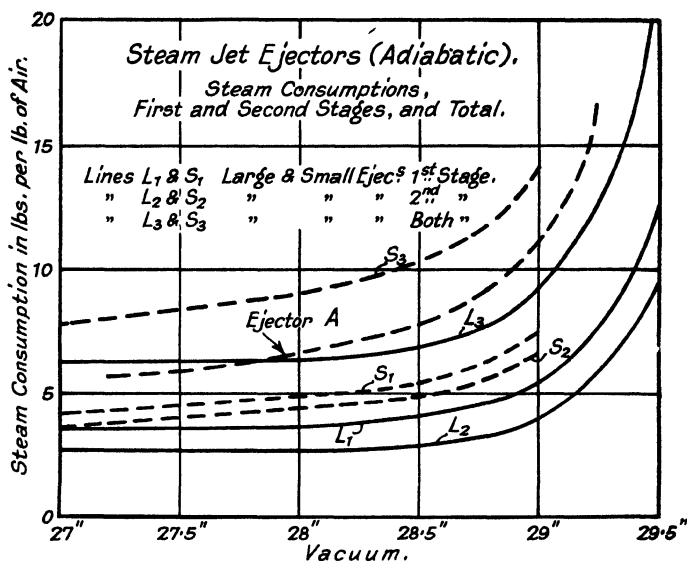


FIG. 206.

GRAPH OF EJECTOR STEAM CONSUMPTIONS

Ejector A is that described on page 332.

In Fig. 207 there is shown the performance of a Ljungström two-stage steam ejector which was supplied with a constant quantity of steam throughout the tests. This weight of steam was about 240 pounds per hour, and from this figure the steam consumption per pound of air at various vacua appears to be—

Vacuum at Ejector Suction Inches	Total Steam per Lb of Dry Air.
29.5"	72.8
29.0"	23.7
28.0"	12.9
27.0"	8.8

These consumptions are higher than those mentioned previously,

but the ejector is quite a small one and higher steam consumptions are to be expected.

Test on a Delas Ejector. There have been published* some details of tests with a water-cooled Delas steam jet ejector. These experiments were made at the works in Paris of the Société des Condenseurs Delas, and the apparatus was arranged as shown diagrammatically in Fig. 208. Air enters the apparatus by the nozzle shown in the upper left-hand corner. Means are provided for heating the

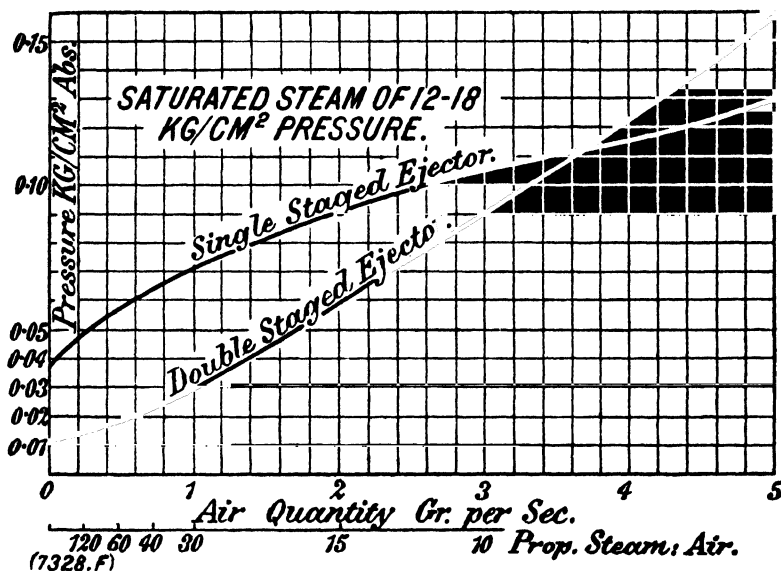


FIG. 207.

PERFORMANCE OF LJUNGSTROM EJECTOR.

injection water which is afterwards removed from the inter-condenser via the hypothetical condenser A.

The results from Test No. 1, using 600 lbs. of superheated steam per hour, are shown in Table XIX, whereas Table XX gives the result of tests with the same injector when using 716 lbs. of nearly saturated steam per hour. In both cases the steam pressure was 242 lbs. per sq. in. gauge. In the former case the average steam temperature was 695° F., and in the latter 412.5° F. From Callandar's *Steam Tables* (Abridged Edition, Fahr. Units) the total heats per pound of steam are 1,217.7 and 1,372.4, heat units respectively. The total amount of heat supplied in the saturated

* *Engineer*, 12th May, 1922 (p. 530).

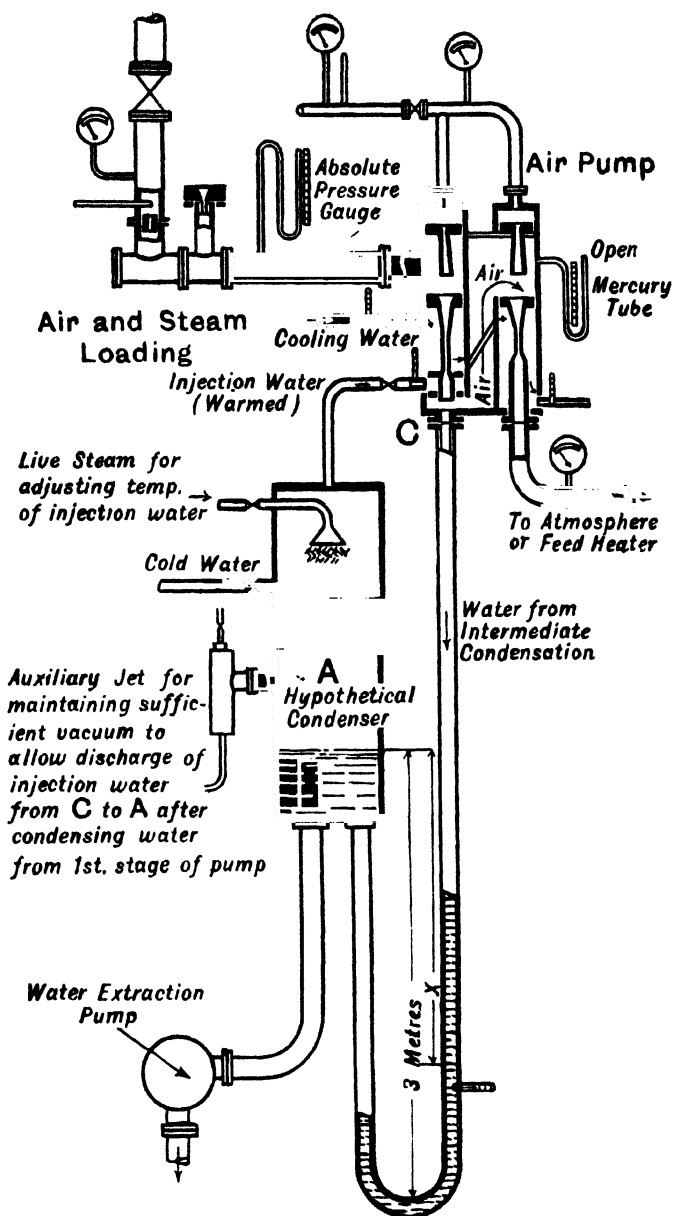


FIG. 208.

DELAS EXPERIMENTAL APPARATUS.

steam tests is about 5.9 per cent greater than in the superheated tests

TABLE XIX
(SUPERHEATED STEAM 600 LBS. PER HOUR)

Air Nozzles. Dia. Inches.	Air Introduced. Lbs. per Hour.	Vacuum. In.	Inter-condenser Vacuum. In.
0	0	29.764	27.76
.079	6.2	29.705	27.84
.118	13.7	29.646	27.68
.157	23.8	29.489	27.29
.197	36.4	29.33	26.76
.236	53.0	28.94	25.80
.275	72.0	28.55	24.70
.275 + .118	85.7	28.31	23.95
.275 + .197	108.4	27.8	22.15
.275 + .236	125.0	26.82	21.20

TABLE XX
(SATURATED STEAM, 716 LBS. PER LB.)

Air Nozzles. Dia. Inches.	Air Introduced. Lbs. per Hour.	Vacuum. In.	Inter-condenser. Press. In.
0	0	29.724	28.27
.029	6.2	29.666	27.68
.118	13.7	29.59	27.41
.157	23.8	29.45	27.02
.197	36.4	29.31	26.58
.236	53.0	29.14	24.19
.225	72.0	28.55	24.90
.275 + .118	85.7	28.38	24.18
.275 + .197	108.4	27.92	23.02
.225 + .236	125.0	27.60	21.92
.275 + .236 { + .118 }	138.7	27.37	20.94

In order to compare the steam consumptions per pound of air with those given on page 330, the air capacities at 29 ins. and 28 ins. are required. These capacities are 58 and 105 lbs., and as the steam consumption is constant at 716 lbs. per hour the consumptions per pound of air are 12.3 and 6.8 lbs. On page 330, the 29 in. consumptions were from 9.2 to 14.0, and the 28 in. from 6.3 to 9.0.

The effect of size of ejector upon consumption is shown clearly

in the tables on page 330, and in order to compare the Delas consumptions with those of a non-cooled ejector, the following figures relating to an ordinary ejector (The Metropolitan-Vickers Electrical Co., Ltd.) taking 770 lbs of saturated steam per hour are given—

Vacuum	Steam per Lb of Air	Air Capacity Lbs per Hour
29.57"	50.0 lbs	15.4
29.41"	25.0 "	30.8
29.25"	16.7 "	46.1
29.10'	12.5 "	61.6
28.90'	10.0 "	77.0
28.60'	8.33 "	92.5
28.30	7.14 "	106.8
27.87"	6.25 "	123.2
27.20	5.55 "	138.8

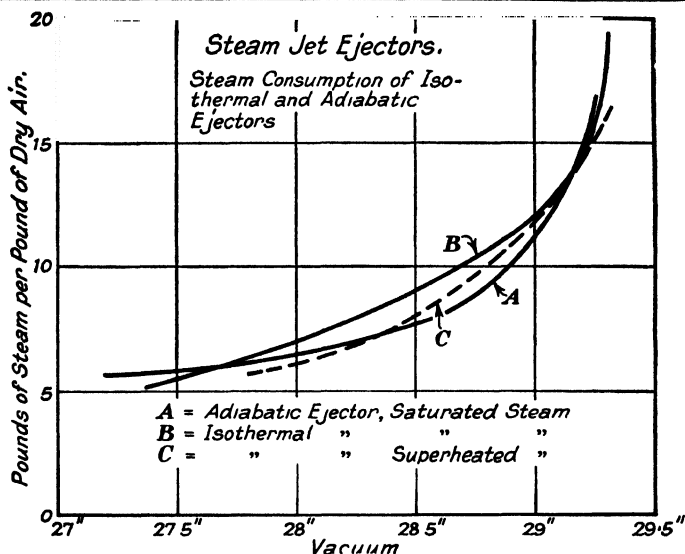


FIG 209

COMPARISON OF STEAM CONSUMPTION OF EJECTORS

These figures and the consumption of steam per pound of air derived from the test details shown on page 334 are graphed in Fig 209. In this figure, line A shows the steam consumption of the ordinary or adiabatic ejector as given above. Line B shows the steam consumption of the isothermal ejector using saturated steam, whilst the lower line C, chain dotted, relates to the use of

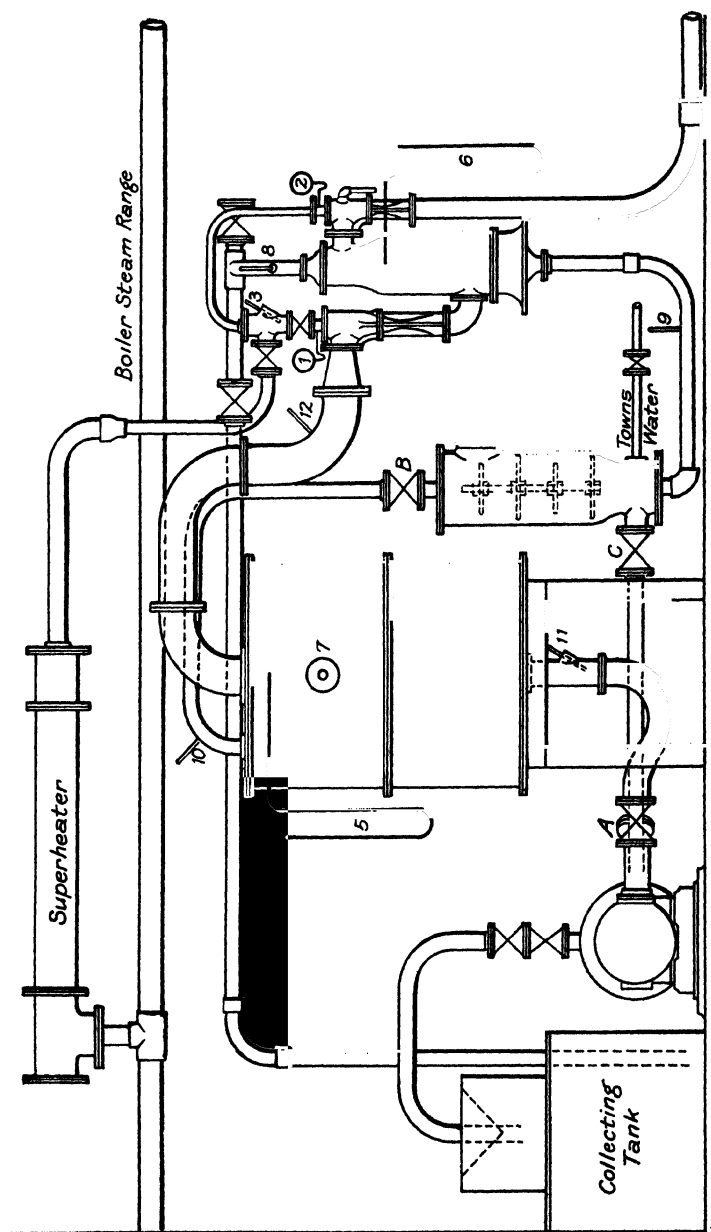


FIG. 210.—ARRANGEMENT OF TEST PLANT.
(Hick, Hargreaves & Co., Ltd.)

superheated steam in the isothermal ejector. This figure, and no other figures are available, shows that the isothermal ejector has no great advantage in consumption within the range of vacua which is more generally used, i.e. 28 ins. to 29 ins.

These two series of tests have been made by different observers, and there may be reasons, not evident in the information available, why corrections should be made to one or to both of them. In any comparison of the two types the tests should be made by the same person who could then ensure that the tests were on the same basis.

The steam consumption of the above adiabatic ejector is shown in Fig. 206, for comparison with the large and small ejectors which were there discussed.

Tests of a Hicks-Breguet Ejector. The next series of tests refer to a Hicks-Breguet air ejector, or "Ejectair," for details of which see Figs. 117 and 118.

The testing plant was arranged as shown diagrammatically in Fig. 210. Steam is supplied through a superheater to the ejector, the temperature being measured at 3, and the first and second stage pressures at 1 and 2. Injection water is aspirated from the tank on the left, the temperature being noted at 8. The water, after leaving the inter-condenser, can pass directly to the pump if valve C is open and valves A and B closed. If C is closed and the other two are open, the water passes into the cylinder shown into which air is admitted through a nozzle at 7. From this cylinder air is withdrawn by the ejector. If the water passes straight to the pump, the air withdrawn from the cylinder is dry, or as nearly dry as the atmospheric conditions will allow, but if the water is passing through the cylinder the air is raised in temperature and becomes as nearly saturated as is possible without admitting steam.

Steam Pressure. Lbs. per sq. in. Gauge.	Per cent Air Duty.	Main Vacuum.	Inter condenser Vacuum.	Inter condenser Inlet Water Temperature.	Suction Air ; Temperature Before Nozzle.
150	0%	29.45	27.7	50° F	59° F.
"	25%	29.35	27.0	54° F	59° F.
"	50%	29.26	26.25	60° F.	59.5° F.
"	75%	29.13	25.5	66° F.	59.5° F.
"	100%	28.87	24.75	74° F.	59.5° F.
"	125%	28.7	23.85	81° F.	59.5° F.
"	150%	28.25	22.85	92° F.	60° F.
"	175%	27.6	21.4	104° F.	60° F.
"	200%	26.5	19.25	118° F.	60° F.

The above results are shown in a graphical form in Fig. 211.

Plant by Worthington-Simpson, Ltd. In Fig. 212 there is shown the arrangement of a 10,000 kw. surface condensing plant by Worthington-Simpson, Ltd., installed in the Radcliffe power station

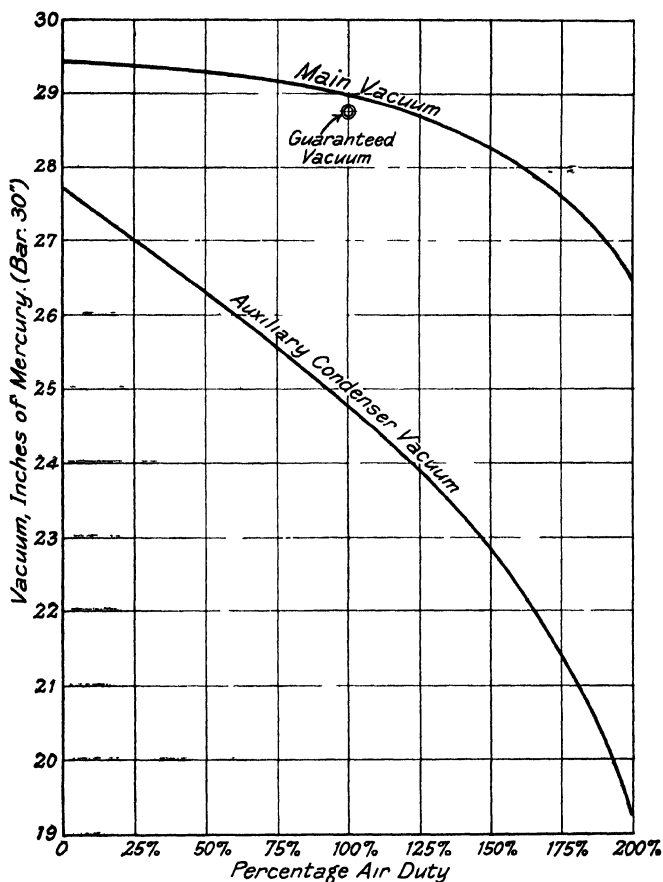


FIG 211
WORKS TEST ON BREGUET EJECTOR.
(Hick Hargreaves.)

belonging to the Lancashire Electric Power Co. It will be seen that there are two circulating pumps, driven by DC and AC motors respectively, and also on the right-hand side of the illustration, two operating pumps supplying water to the hydraulic air ejector. For details of this particular ejector, see pages 277 to 280. The air suction from the condenser will be seen to the right of the condenser,

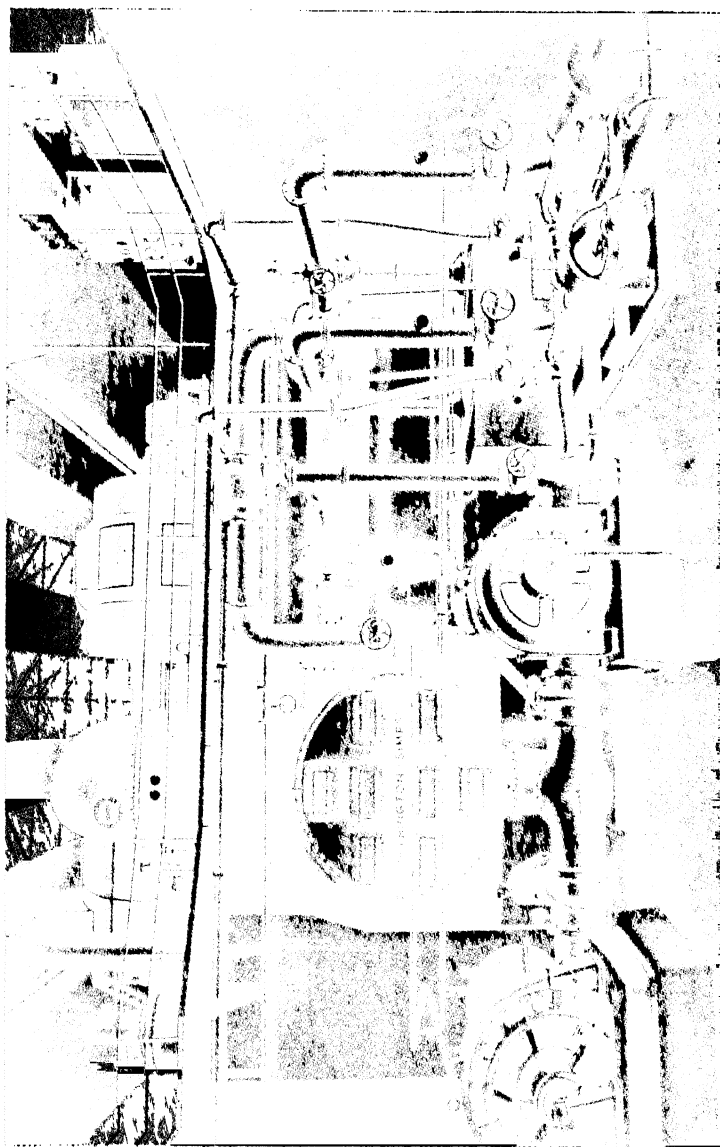


FIG 212
ARRANGEMENT OF A 10,000 KW CONDENSING PLANT.
(Worthington Simpson Ltd)

fitted with sluice and non-return valves. Each operating pump has a condensate extraction pump coupled to it, and the necessary pipe connections can be seen. This plant was guaranteed to maintain a vacuum of 28.5 ins. (barometer 30 ins.) when dealing with 120,000 lbs. of steam per hour, and supplied with water at a temperature of 65° F. The surface was 20,000 sq. ft., and each pumping unit was designed to deal with 60 per cent of the full load. This condenser is fully described on pages 83-85.

The average of the various readings at the different tests are given in the following table—

Test No.	Lbs. of Steam per Hour.	Keno meter Readings.	Exhaust Steam Temp. ° F.	Circulating Water.		Operating Water Tank Temp. ° F.	Air Pump Vapour Temp. ° F.	Extraction Suction Temp. ° F.
				Inlet. ° F.	Outlet. ° F.			
1	125,438	1.003"	—	60.9	74.3	65.8	68.8	76.2
2	92,500	0.948"	—	61.0	71.2	66.0	68.5	69.0
3	61,228	0.945"	—	61.6	71.6	65.8	69.7	67.1
4	154,678	1.220"	86.5	62.9	78.9	68.3	71.6	81.7
5	128,300	1.457"	91.5	63.0	85.6	69.0	75.5	88.1

Tests Nos. 1 and 5 are both on full load, but in the latter, only one circulating pump was in use. In the first three tests it is stated that the exhaust steam temperatures were within 1° of those corresponding to the recorded vacua. In tests Nos. 4 and 5, the recorded temperatures are 1° above those corresponding to the vacua, and assuming that the same relation holds for the other tests then the transmission coefficient K (B.Th.U. per square foot, per hour, per degree, mean difference of temperature) is as follows—

Test No.	K
1	566
2	405
3	281
4	561
5	461

Plant by Richardsons, Westgarth & Co., Ltd. Very complete tests have been made upon a condensing plant supplied by Richardsons, Westgarth & Co., Ltd., to the Manchester Corporation Electricity Works at Stuart Street in connection with a 8,000/10,000 kw. turbine. The condenser is of the Contraflo type with a surface of 15,000 sq. ft., and is fitted with a two-stage steam ejector with inter-condenser. The test data are too numerous to be reproduced in full, but certain points will be examined in detail.

If an air pump were to be supplied apart from a condenser, its performance would be specified by the partial air pressure which

it could maintain in the condenser when handling a definite weight of air and with water at a specified temperature in the condenser. It therefore is interesting to ascertain the actual partial air pressure maintained in this plant.

In all eight tests were made at loads between 4,389 kw. and 9,713 kw., and the following figures refer to the absolute pressures and temperatures at the air pump suction on the condenser—

	Test No.					
	1	2	3	4	5	6
Temperature at air pump suction—°F.	95.3	94.7	91.7	96.0	88.1	79.5
Corresponding vapour pressure—In.	1.678	1.647	1.501	1.714	1.340	1.017
Actual total pressure at air pump suction	1.72	1.67	1.51	1.75	1.35	1.10
Partial air pressure—In.	0.042	0.023	0.009	0.036	0.010	0.028
Steam load—Lbs. per hour	96,400	98,900	76,080	115,020	54,300	96,466

This shows that the air pressure is extremely low, the air extractor being of ample capacity for the duty and the air leakage very small. The air pressures for tests Nos. 3 and 5, are much below the others. It should be noted that if the air pump suction temperatures in those tests had been incorrectly read and should be even 0.5° F. lower, the air partial pressure as calculated above would have been .032 ins. and .030 ins. respectively. As the calculations are so sensitive to small errors in the observations, the results cannot be expected to be proportional to the steam load, but they show that the air extractor is highly efficient.

When a similar calculation is made regarding the air partial pressure at the top of the condenser the results are more variable. The absolute pressure was measured by two mercury columns and also by two absolute pressure gauges (Kenotometers). The two averages of these two methods of observation differ as will be seen below. If the mercury column readings be taken, the actual absolute pressure, including air as well as steam, is less than the vapour pressure corresponding to the recorded temperature.

	Test No.							
	1	2	3	4	5	6	7	8
Temperature at condenser top —°F.	97.9	97.2	94.1	99.4	89.6	83.6	98.6	95.3
Corresponding vapour pressure—In.	1.817	1.778	1.616	1.899	1.406	1.168	1.855	1.678
Absolute Pressure by—								
(1) Columns—In.	1.66	1.72	1.56	1.79	1.35	1.17	—	—
(2) Kenotometers—In.	1.84	1.82	1.67	1.89	1.37	1.20	1.97	1.78
Air Pressure by—								
(1) Columns—In.	—	—	—	—	—	0.007	—	—
(2) Kenotometers—In.	0.023	0.042	0.054	—	—	0.037	0.115	0.102

TEMPERATURE CORRECTIONS OF MERCURY COLUMN READINGS. Dealing solely with the above mercury column readings, it should be noted that the recorded absolute pressures at the condenser top were measured by a column in a warm situation and no correction has been made for temperature. Assume that the temperature of the column was 95° F. (which is probably on the high side). In standard English barometry the mercury column is reduced to 32° F., and therefore the observed heights of the columns should be reduced to this temperature. If the barometric readings made during the tests have not been reduced to 32° F., the mercury column readings should at least be reduced to the temperature at which the barometric readings were made. If the latter temperature be assumed to be 59° F., the height of the mercury column must be reduced in the ratio of the densities of mercury at 95° and 59° F. or $13.5095/13.5584 = 0.9964$. This is a reduction of approximately 1/3 per cent, and applied to a column reading of 28.34 ins. (Test No. 1) gives a corrected reading of practically 28.24 ins., thus increasing the absolute pressure from 1.66 ins. to 1.76 ins./Hg. If the barometric readings were corrected to 32° F., then the correction to the column readings would be 0.63 per cent, and the absolute pressure increased from 1.66 ins. to 1.84 ins. Unless, therefore, the average temperature of the mercury column is recorded accurately, information as to the partial air pressure either at the top or bottom of the condenser cannot be correctly deduced.

At this point it may be of interest to include the factors for the correction of mercury column readings to the equivalent reading at 0° C. Working from the accepted densities of mercury* the percentage reductions in the observed readings are—

Temperature.		Reduction in Mercury Column Reading.
Degrees Centigrade.	Degrees Fahrenheit.	
0	32	Nil.
20	68	0.3636%
40	104	0.7223%
60	140	1.0805%
80	176	1.4370%
100	212	1.7925%

The percentage reductions are obtained with sufficient accuracy from the expression

$0.017925 t^{\circ}$ where t is the mercury temperature in degrees centigrade, or

$0.00995 (t - 32)^{\circ}$ where t is in degrees fahrenheit.

* Kaye and Laby. *Physical and Chemical Constants*.

The latter may be taken as 0.01 ($t-32$) per cent with an accuracy as great as that of the observations.

VARIATION OF PRESSURE AT STEAM INLET TO CONDENSER. It was stated above that the Kenotometer readings tabulated on page 341 were the average of the readings of two instruments. Each was connected to a different side of the turbine exhaust flange, and the difference between the two readings varied from 0.10 in. at

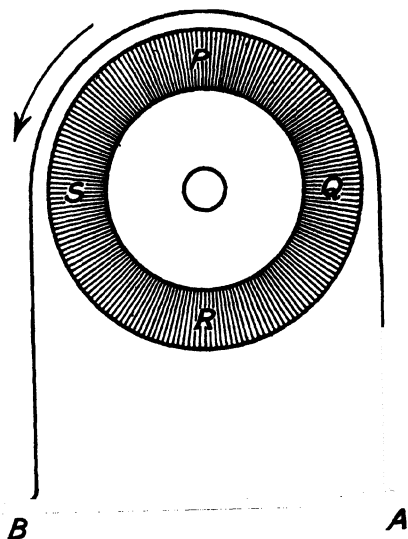


FIG. 213.

SECTION THROUGH TURBINE EXHAUST.

overload down to 0.03 in. at one $\frac{3}{4}$ load test, but was not proportional to the load. Let Fig. 213 represent diagrammatically a section through a turbine exhaust. If the steam left the blades in an axial direction the pressure at the exhaust flange would be uniform across the flange in a transverse direction, but a lesser pressure would be expected at the part remote from the last row of blades than at the part nearest to those blades. When the steam no longer leaves the blades axially it becomes heaped at one side. Thus suppose the rotor to revolve counter clockwise and the steam to flow back over or to have a clockwise component as well as the axial component. The steam leaving at P will flow towards the walls at side A. Steam leaving at Q will tend to distribute evenly over the exhaust flange at side A. The steam leaving at R will have a motion towards the wall at side B, but will also be drawn

downwards towards the condenser. The steam leaving at S will have an initial velocity upwards which is overcome by the condenser pull. Thus the steam will be moving more rapidly at side A than at side B, and the absolute pressure should therefore be greater at B than at A. In order that this condition may be stable, the average direction and velocity of the steam flowing from the last row of blades to the exhaust flange must be such as to have a horizontal component corresponding to this pressure difference. With a vacuum of 28 ins. and a pressure difference of 0.1 in. or 0.049 lb. per sq. in., the necessary velocity is obtained from the simplified flow of gas equation

$$v^2 = \frac{2g}{w_0} (p_0 - p)$$

In this case, $v = 392$ feet per second. Assuming that the average velocity of the exhaust steam at the exhaust flange is 400 feet per second, the average direction of flow is inclined at an angle of $\tan^{-1} \frac{400}{392}$ or 45° approximately. Flow in this direction cannot be maintained, due to the action of the side A, and so the pressure across the passage to the condenser tends to become equalized, but with the usual short connections between turbine and condenser, the pressure is probably never uniform across the whole passage.

METHOD OF EQUALIZING PRESSURE AT STEAM INLET TO CONDENSER. Some information relating to this point has been published by H. L. Guy and P. L. Jones.* They used a special diffuser with the object of spreading the steam uniformly over the whole length of the tubes and of avoiding the inequality of pressure. Fig. 214 shows a sectional elevation of a marine turbine and condenser with the diffuser inserted. The diffuser section is shown separately in Fig. 215. Actual tests with a condenser having an inlet 16 ft. $\times$ 6 ft. gave the following results. The steam passed was 228,310 lbs. per hour, and the maximum vacuum measured just above the turbine exhaust flange was 28.62 ins. Assuming that the steam was dry and saturated and that the average vacuum did not vary greatly from this figure, the mean velocity of the exhaust steam was approximately 318 feet per second. The maximum vacuum or minimum absolute pressure was obtained at the point indicated in Fig. 216. The pressure differences between this point and other points were measured by a differential gauge, using paraffin, and are shown in the illustration. It will be seen

* *Japanese Society of Naval Architecture*, October, 1922, and *Engineering*, February, 1923.

that the pressure is higher at the side of the exhaust remote from the last wheel of the turbine and also that it is greater on the right-hand



FIG. 215.
TURBINE EXHAUST DIFFUSER.
(Metropolitan-Vickers Electrical Co., Ltd.)

side than at the left, showing that there must be a horizontal component of the velocity from the right to the left. Only one observa-

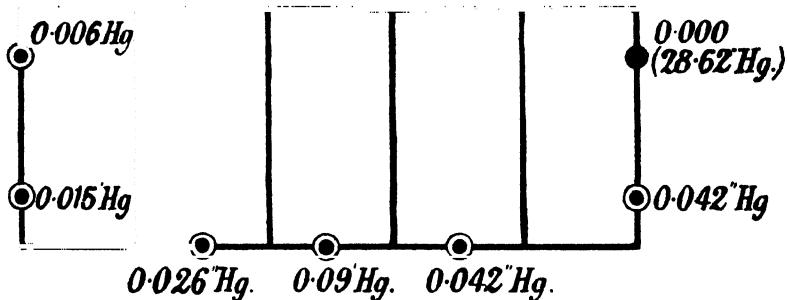


FIG. 216.
PRESSURE DIFFERENCES AROUND EXHAUST FLANGE OBTAINED BY
MEANS OF PARAFFIN DIFFERENTIAL GAUGES.

tion is recorded in the space between each pair of diffuser vanes, and it would have been interesting to test the uniformity of pressure across each of these spaces.

Air Quantities. The Manchester tests enable approximate calculations to be made regarding the quantity of air handled by the extractors. Taking test No. 1 (page 365) as an example, the following are the data observed in connection with the two-stage ejector.

Vacuum in condenser at ejector suction	28.28 in.
Steam pressure to first stage	45 lbs. per sq. in. g.
Steam temperature to first stage	635° F.
Inter-condenser vacuum	27.86 in.
Temperature at ejector suction	95.3° F.
Steam pressure to second stage	145 lbs. per sq. in. g.
Steam temperature to second stage	635° F.
Steam supply to first stage (calculated)	183 lbs. per hour.
Steam supply to second stage (calculated)	802 lbs. per hour.

It has been shown that the efficiency of an adiabatic steam-jet ejector is about 4.0 per cent (page 207) and the method of calculation is to ascertain the energy supplied to the first stage, assume an efficiency of 4 per cent, calculate the work done per pound of mixture, and so ascertain approximately the weight of mixture and hence the weight of air.

In the above figures two points are noticeable, first the very low steam pressure at the first stage, 45 lbs. g., and second the small pressure increment or plenum in the first stage of 0.42 in. These figures indicate that the quantity of air is very small as it has been shown that with no air the increment in the first stage is nil.

Steam expanding adiabatically from 45 lbs. gauge or 60 lbs. per sq. in. absolute to a vacuum of 27.86 ins. has an available heat drop of 323.45 B.Th.U.'s. The supply of steam was 183 lbs. per hour, calculated by the Rateau formula, and if the ejector efficiency is 4 per cent, then the energy actually utilized to compress the mixture of air and vapour is,

$$323.45 \times 183 \times 777.8 \times 0.04 = 1,841,000 \text{ ft.-lbs. per hour.}$$

The mixture is compressed from an initial vacuum of 28.28 ins. to 27.86 ins., or 0.842 to 1.05 lbs. per sq. in. The vapour pressure at 95.3° is 1.678 ins., and as the total absolute pressure is 1.72 ins. the partial air pressure is .042 ins. or .0206 lb. per sq. in. absolute. The density of steam at 95.3° is .0025 lb. per cu. ft., and of air at this temperature and a partial pressure of .0206 lb. per sq. in., .0001006 lb. per cu. ft., and the specific volume 384.5 cu. ft. per lb.

Substituting these values in the equation

$$\frac{p_1 v_1}{\gamma - 1} \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right]$$

and assuming that γ for the mixture is 1.40, the work done in compressing one pound of the mixture is 7,575 ft.-lbs. Hence the weight of mixture extracted per hour is

$$\frac{1,841,000}{7,575} = 243.1 \text{ lbs.}$$

The weight of air is

$$243.1 \times \frac{.0001006}{.0026006} = 9.35 \text{ lbs. per hour.}$$

Similar calculations could be made for the second stage if the temperature at which the air reaches the second stage steam jet after passing through the inter-condenser were known. Condensate was supplied from the extraction pump discharge at a temperature of 98.15° F., and left the inter-condenser at 106.0°. The air temperature would be a little above the condensate inlet temperature. If it be assumed to be 100° and alternatively 101°, similar calculations show the air per hour to be 11.24 and 7.95 lbs. respectively, so that a temperature of 100.6° F. would correspond to 9.35 lbs. of air per hour, the amount obtained above from the first stage particulars.

This calculation can only be regarded as approximate for the reasons detailed earlier when considering the accuracy of mercury column readings, and also on account of the assumed efficiency of the steam ejector. In this case the vacuum after the first stage was read by a calibrated Bourdon gauge.

Recommended Air Capacity for Various Plants. The steam load in this test was 96,400 lbs. per hour, so that the calculated air is equal to 0.97 lb. per 10,000 lbs. of steam. This figure is considerably below the quantities usually assumed in designing plants. For each 10,000 lbs. of steam, at full load, designers will usually allow for 5 lbs. of air, but in doing this they feel that with a normal air-tight plant they have a 100 per cent margin, i.e. that the air should not exceed 2.5 lbs. This latter figure is supported by Kothny,* who stated that investigations have shown that the normal air leakage for marine installations can be limited to a quantity varying from 7.5 lbs. at zero steam load to 32.5 lbs. per hour with a steam load of 100,000 lbs., or an increment at the rate of 2.5 lbs. per 10,000 lbs. The zero amount of 7.5 may be needed for marine plants, but for good land plants this figure can probably be reduced by 50 per cent to 60 per cent.

The Wheeler Condenser & Engineering Co. have published a curve

* *American Society of Naval Architects*, New York, November, 1919.

giving their recommendations as to the air handling capacity to be provided in plants of various size. When converted from a volume basis this curve is equivalent to 6.9 lbs. per 10,000 lbs. with a steam capacity of 100,000 lbs. per hour, falling to 4.6 lbs. with a capacity of 350,000 lbs. per hour.

The British Electrical & Allied Manufacturers Association (BEAMA) recommend that the designed capacity of air ejectors be calculated by the empirical formula—

$$\text{Dry Air capacity} = \left(\frac{\text{steam condensed per hour}}{2,000} + 3. \right) \text{ lbs. per hr.}$$

Further information as to the quantity of air handled by an air pump has been published in connection with the tests on a 30,000 kw. steam turbine* by Herbert B. Reynolds. The air pumps were of the two-stage dry vacuum type, and the air discharged was measured in a gasometer, thus affording a means of direct observation. The data recorded is as follows—

Steam Load. Lbs. per Hour.	Air Discharged in Cubic Feet per Minute at 60° F.
180,666	9.94
317,000	9.47
373,666	6.50

The specific volume of air at 60° F. being 13.06 cu. ft., the weight of air per hour for the three cases was 45.6, 43.5 and 29.9 lbs. Dividing by the steam loads, these figures are equivalent to 2.52, 1.37, and 0.80 lbs. per 10,000 lbs. of steam.

Similar tests were made by the same engineer on a 60,000 kw. triple-cylinder cross compound turbine. On the maximum load of 60,233 kw., with a steam consumption of 725,667 lbs. per hour, the air discharge was 14.8 cu. ft. per minute at 60° F. This is equivalent to 68 lbs. of air per hour, or 0.936 lbs. per 10,000 lbs. of steam.

The Manchester test gave 0.97 lb. per 10,000 lbs., so that these figures confirm each other as showing what may be expected in an air-tight plant. It is interesting to note that the total weight of air handled decreased with increasing steam load, this probably indicating that with the higher load, the pressure so increased at one or more points as to reduce or prevent the ingress of air.

* *Proceedings American Society of Mechanical Engineers* (Vol. XLIII, p. 130).

MANCHESTER TESTS. VALUE OF K. The Manchester test shows that the plant is particularly air-tight, and that the air extractor is able to maintain a very low partial air pressure at the air suction and, hence, throughout the whole condenser.

The coefficient (K) has been worked out from the following data. The cooling surface was 15,000 sq. ft.

	Test No.					
	1	2	3	4	5	6
Circulating water inlet, °F.	79.9	80.2	79.58	77.94	79.4	66.06
" " outlet, °F.	93.9	93.54	90.9	94.7	87.6	80.31
Top of condenser, °F.	97.9	97.2	94.1	99.4	89.6	83.6
Circulating water, G P H	653,250	649,300	637,000	658,060	622,550	643,250
K	654	664	640	667	677	720

In calculating K, the logarithmic mean temperature difference has been used, and the steam temperature has been assumed to be constant and equal to the temperature at the top of the condenser. Actually in test No. 1, the steam temperature dropped 2.6°, and if this were taken into account the mean temperature difference would be reduced from 10.4° to 9.32° and K increased from 654 to 730, an increase of 11.5 per cent.

Plant by C. A. Parsons & Co., Ltd. An example of a high value of K in ordinary commercial practice is provided by the following partial load figures relating to a 4,000 kw. plant by C. A. Parsons & Co., Ltd., at Redditch—

Surface	4,500 sq. ft.
Steam load	17,060 lbs. per hour.
Vacuum (Bar. 30")	29.1 in.
Circulating water inlet	66.25° F.
Circulating water outlet	73.35° F.
Vacuum temperature	76.2° F.
K	667

Plant by Alberger Condenser Co. The following tests,* shown on the next page, relate to Alberger condensers fitted with the makers' own steam ejectors.

The above values of K, which are unusually high for regular practice are calculated upon the heat transfer as indicated by the rise in temperature of the circulating water. As both the steam consumption and the quantity of circulating water is estimated,

	Date of Test.	
	26th Aug., 1919.	4th April, 1922.
Capacity of turbine	10,000 kw.	8,000 kw.
Surface of condenser	19,400 sq. ft.	15,000 sq. ft.
Test load	8,400 kw	6,500 kw.
Steam load (estimated)	97,500 lbs. per hr	83,900 lbs. per hr
Absolute pressure at condenser inlet	1.312 in./Hg.	0.95 in./Hg.
Corresponding temperature	87.4° F	77.4° F.
Circulating water inlet	76° F.	60° F.
Circulating water outlet	83.5° F	75° F.
Circulating water (estimated)	24,600 G P M.	10,620 G.P.M.
Mean temperature differences (Logarithmic)	6.55° F	7.58° F.
K	857	840

the above values of K may not be correct. The small temperature differences between the circulating water outlet and the steam inlet, 3.9° F. and 2.4° F. show, however, that the condensers are putting up good performances, particularly when it is considered that neither are fully loaded. No information is available as to the means adopted for measuring the absolute pressures. If a mercury column was used, the reading should have been corrected to 32° F., thus increasing the recorded absolute pressure.

Plant by Wheeler Condenser & Engineering Co., Ltd. The following figures* relate to a 18,000 sq. ft. plant by the Wheeler Condenser & Engineering Co., at El Paso, Texas, using artificially cooled water—

Steam per hour	105,500 lbs	102,800 lbs.
Vacuum (Bar. 30")	27.95 in./Hg	28.00 in./Hg.
Circulating water inlet	82° F.	82° F.
Circulating water outlet	95° F.	94° F.
Temperature corresponding to vacuum	102° F.	101° F.
K	472	474

Plant by The Mirrlees-Watson Co., Ltd. A typical modern condensing plant of medium size is that supplied by Mirrlees-Watson, Ltd., to the Glasgow Corporation for the new Dalmarnock power station. This consists of a wedge-shape condenser with a surface of 26,000 sq. ft., the general arrangement being illustrated in Fig. 217. When

* *Report of Prime Movers Committee, N.E.L.A., 1922.*

supplied with 22,500 gals, per minute of circulating water at a temperature of 47° F., the following results were obtained—

Steam Load. Lbs. per Hour.	Vacuum on Bar. 30".	Circulating Water Outlet.	Condensate Temperature.
123,000	29.46 in.	57.0° F.	53° F.
143,000	29.30 in.	55.8° F.	55° F.
180,000	29.30 in.	61.0° F.	56° F.

On the above figures the values of K were 606, 461, and 531 respectively. The circulating water outlet temperatures are stated as recorded but appear to be inconsistent.

Plant by Schutte & Koerting. Table XXI gives details of a test conducted on a No. 41½ multi-jet ejector condenser by Schutte and Koerting of Philadelphia. This condenser is rated at 70,000 lbs. of steam per hr. at 28 ins. vacuum and 70° F. water. Although the test was conducted in the summer and the injection water temperature varied from 70° to 79° F., the vacuum referred to a 30-inch barometer was maintained well above 28 ins. The condenser tested served a 3,000 kw. G. E. Curtis turbine, and the test was run under commercial conditions. These details clearly confirm

TABLE XXI

PERFORMANCE TEST ON NO. 41½ KOERTING LOW-LEVEL MULTI-JET CONDENSER AT WHITAKER-GLESSNER CO

DATE OF TEST 1917.				
Item	20th Aug	20th Aug	22th Aug.	25th Aug
Barometer observed . . . in	29.41	29.41	29.32	29.37
Vacuum observed . . . in.	27.75	27.7	27.94	27.5
Absolute pressure . . . in.	1.66	1.71	1.38	1.87
Vacuum referred to 30-in. Barometer . . . in	28.34	28.29	28.62	28.13
Injection water temperature . °F.	79	78	78	76
Hot well temperature . . °F.	88	83	84	88
Vapour tension corresponding to hot well temperature . . in.	1.334	1.137	1.174	1.334
Vacuum theoretical . . . in.	28.666	28.863	28.826	28.668
Air tension . . . in.	0.326	.573	.206	.536
Percentage of theoretical vacuum obtaining . . . %	98.7	98.0	99.3	98.0
Water pressure . . lbs. per sq. in.	0	9	8	10.5
Water . . . g.p.m.	5,520	7,100	7,000	7,300
Steam . . . lbs. per hr.	28,300	16,850	20,000	41,800
B.T.U.. . . per lb. steam	1045.4	1050.8	1046.7	1047.2

the fact that in jet condensers the partial air pressure is much greater than in surface plants due to the air introduced with the injection water.

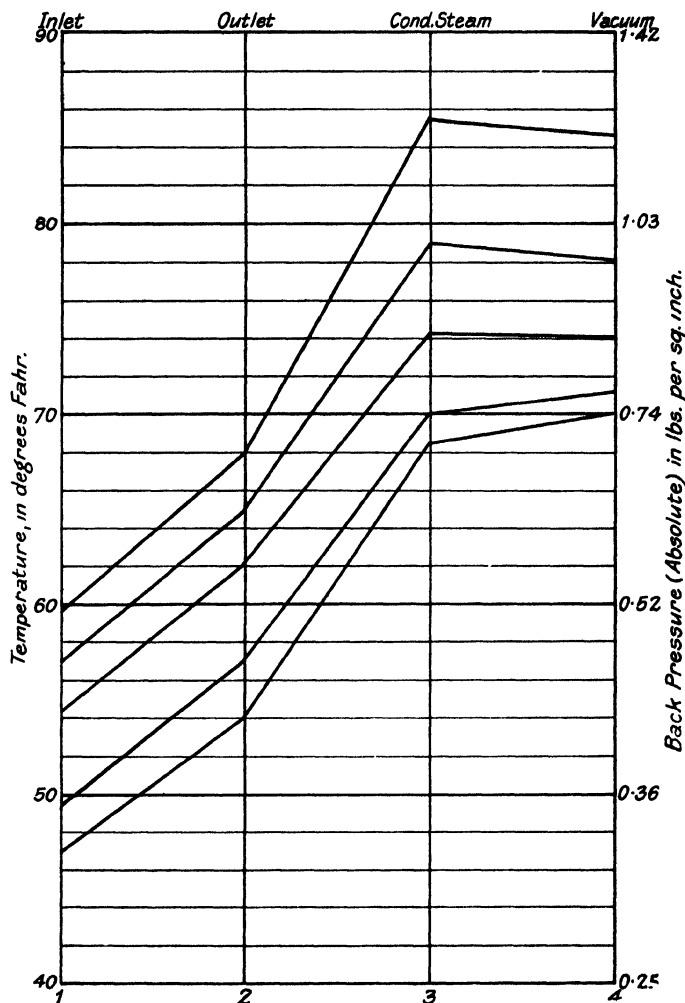


FIG. 218.

CONDENSER OPERATING CHART.

Method of Recording Routine Readings. Although not directly connected with tests of condensing plants, reference may be made here to an interesting method of recording routine readings of

plant performances, particularly when the temperature of the circulating water temperature varies. Such a case occurs in a station dependent upon cooling towers, and where the towers are rather small for the peak load. In Fig. 218 there is reproduced an illustration which appeared in the Prime Movers Committee Report of the National Electric Light Association for 1922. The vertical lines 1, 2 and 3, are equally divided temperature scales, and line 4 is graduated in absolute pressure, the pressure at any point on this line being that corresponding to the temperature at the same level on lines 1, 2 and 3. On line 1 there is recorded the circulating water inlet temperature, on line 2 the outlet temperature, on line 3 the condensate temperature, and on line 4 the condenser absolute pressure. Assuming that any given slope of line in each section represents normal performance, then a change in slope indicates some change in the operating conditions. As stated by the Committee, if the slope of the line is steeper in the first section of the curve it indicates that insufficient water is supplied, and if the slope is steeper in the second section, that the tubes are dirty, and if the actual back pressure is higher relatively to the condensed steam temperature it indicates that the air removal system is less efficient.

CHAPTER XXVI

CLOSED FEED SYSTEMS AND DE-AERATING PLANTS

ALL water contains air in solution. The solubility decreases with a rise in temperature, a fact demonstrated every time a pan of water is boiled. As the temperature is raised the air rises to the surface and escapes, and when the water is actually boiling the air contents of the water is nil.

In a boiler supplying steam to a power plant fitted with the usual arrangement of feed system, air is constantly introduced into the system partially by absorption from the atmosphere and mainly by direct leakage at points of the system where the pressure falls below the atmospheric. This air is carried forward with the steam, and at the entrance to the condenser the air is present probably in the ratio of one part of air to 2,500 or 3,000 parts of steam. On its passage through the condenser the bulk of the steam is condensed leaving the residual mixture relatively richer in air, and the ratio may become one of air to four of steam. This rich mixture is then taken by the air extractor and rejected, the condensate as delivered by the extraction pump containing a very small amount of air. If it could be returned to the boiler in this condition, assuming that there are no air in-leaks at glands, etc., the air to be removed by the air pump would arise entirely from the make-up feed water.

But it is usual for the extraction or lift pump to deliver into an open hot well, whence the boiler feed pump draws. Water exposed to air rapidly absorbs sufficient air to make it air-saturated at its particular temperature. Hence with an open hot well, which is needed to provide surge capacity between the lift and boiler feed pumps, the boiler feed is fully saturated and there is a constant circulation of a relatively large amount of air. The air enters by the hot well and leaves by the extractor.

Effect of Air in Steam. As air has a twofold detrimental effect—

(a) upon the condenser performance, that is, a physical effect, and

(b) upon the economizer tubes, and to a lesser degree on the boiler tubes, that is, a chemical effect,

it is most important to reduce it as much as possible.

The effect of air upon the heat transmission has already been

examined in connection with main condensers, and mentioned with regard to inter-condensers, so that no more need be said here. This drawback has to be countered by providing greater surface with consequently increased cost and capital charges, and its elimination is highly desirable.

As to the economizers, it has now become necessary with the higher steam pressures, which are general, to use steel tubes instead of the usual cast-iron, and experience has shown that these steel tubes are very susceptible to pitting and erosion. It has been proved that this damage is caused by oxygen held in solution, hence it is still more desirable to remove the oxygen.

Air Contents of Aerated Water. In considering any closed feed systems it is useful to know what is the minimum amount of air that is contained in the condensate as it leaves the bottom of the condenser, as this is the condition in which it is desired to retain the condensate until it is delivered to the boiler. Gases are absorbed by water to a greater extent at low temperature than at high, and at the same temperatures the coefficient of absorption varies with different gases. This coefficient may be defined as the volume of gas, measured at 0° C. and 760 mm., which a unit volume of the pure solvent will dissolve when the pressure of the gas without the vapour tension of the solvent is equal to 760 mm. The values for oxygen and nitrogen are given for various temperatures below—

TABLE XXII
COEFFICIENTS OF ABSORPTION OF GASES

Temperature	Oxygen.	Nitrogen.
0° C.	·0489	·0235
5° C.	·0429	·0208
10° C.	·0380	·0186
15° C.	·0342	·0168
20° C.	·0310	·0154
25° C.	·0283	·0143
30° C.	·0261	·0134
40° C.	·0231	·0118
50° C.	·0209	·0109
60° C.	·0195	·0102
70° C.	·0183	
80° C.	·0176	·0096
90° C.	·0172	
100° C.	·0170	·0095

The above figures are taken from Seidell's *Solubilities*.* All data available relating to this subject are based on the centigrade system,

* *Solubilities of Inorganic and Organic Substances*, by A. Seidell, Ph.D., 1920. Crosby, Lockwood.

and for the convenience of those used to the Fahrenheit scale the final figures are given later on that scale. Taking as an example the nitrogen coefficient at 30° C. (86° F.) the weight of N contained in 10,000 lbs. of water may be calculated, assuming that the water is in contact with N only at atmospheric pressure. The weight dissolved is dependent on the pressure of the gas as the solvent will take a certain volume at any temperature, this volume being independent of the pressure. The density of nitrogen at 0° C. and 760 mm. is .0012507 grammes per c.c.* With water at 30° C., the vapour pressure is 0.616 lb. per sq. in., so that the actual N pressure is 14.074 lbs. The density of the nitrogen is reduced in the ratio of 14.074/14.69, and becomes .001199. At 30° C. the N absorption coefficient is .0134, so that the weight of N dissolved by 10,000 pounds of water is

$$.001199 \times .0134 \times 10,000 = 0.161 \text{ lb.}$$

That this calculation is correct is seen from the fact that a coefficient of absorption of say x c.c. of gas per c.c. of water means x c.c. of gas per gramme of water. If s be the specific density of the gas, then xs grammes of gas are absorbed per gramme of water, hence xs lbs. are absorbed per pound of water.

By this means the following figures have been obtained, giving the weights of O and N in 10,000 lbs. of water at various temperatures, the pressure of the gas plus the vapour tension of the water being equal to the atmospheric pressure of 14.69 lbs. per sq. in.

TABLE XXIII
POUNDS OF GAS DISSOLVED PER 10,000 LBS. OF WATER
(Gas Pressure plus Vapour Pressure = 14.69 lbs. sq. inch.)

Temperature.	Oxygen.	Nitrogen.
0° C.	0.695	0.292
5° C.	0.607	0.259
10° C.	0.537	0.230
15° C.	0.480	0.208
20° C.	0.434	0.189
25° C.	0.393	0.174
30° C.	0.359	0.161
40° C.	0.308	0.139
50° C.	0.266	0.121
60° C.	0.227	0.105
70° C.	0.186	
80° C.	0.138	0.069
90° C.	0.079	
100° C.	Nil	Nil

* For such data see *Physical and Chemical Constants*, by Kaye and Laby. Fourth Edition, 1921.

Of the atmospheric pressure, approximately 21 per cent is due to O and 79 per cent to N (including with N, the other inert gases), so that if the above figures for O and N be multiplied by 0.21 and 0.79 respectively the resulting figures will give the weight of O and N absorbed by 10,000 lbs. of water from the atmosphere under normal atmospheric pressure.

TABLE XXIV

POUNDS OF ATMOSPHERIC AIR ABSORBED PER 10,000 LBS. OF WATER

(Air Pressure plus Vapour Pressure = 14.69 lbs. per sq. in.)

Temperature	Oxygen.	Nitrogen.	Total.
0° C.	0.146	0.242	0.388
5° C.	0.127	0.205	0.332
10° C.	0.113	0.182	0.295
15° C.	0.101	0.164	0.265
20° C.	0.091	0.149	0.240
25° C.	0.082	0.137	0.219
30° C.	0.075	0.127	0.202
40° C.	0.065	0.110	0.175
50° C.	0.056	0.096	0.152
60° C.	0.048	0.083	0.131
70° C.	0.039	0.069	0.108
80° C.	0.029	0.054	0.083
90° C.	0.016	0.033	0.049
100° C.	Nil	Nil	Nil

THE SURFACE CONDENSER AS DE-AERATOR. The above figures show that the maximum amount of air contained in 10,000 lbs. of water at the usual condensate temperature, even if subject to the full atmospheric pressure, is quite small, being 0.202 lbs. at 30° C. (86° F.). In a condenser at the air pump suction flange the partial air pressure is only about 0.1 in. Hg., and may be as high as 0.3 in./Hg. This latter figure is only 1 per cent of the atmospheric pressure, so that the weight of contained air at 30° C. will be reduced from 0.202 to 0.00202 lbs. per 10,000 lbs. Thus the amount of air required to saturate water under the conditions prevailing in the condenser is negligible in comparison to the amount which has to be extracted by the air pump.

It is shown on pages 346 to 348, that even in plants which are considered to be very air-tight, that the weight of air to be handled by the air pump is between 0.8 and 1.0 lbs. per 10,000 lbs. of steam.

This consideration shows that the surface condenser is a good de-aerator. Small though the quantity of contained oxygen may be in the condensate, it still facilitates pitting of the economizer tubes, and processes have been introduced with the object of

attaining a further reduction of the oxygen contents of the condensate.

For convenience of those using the Fahrenheit scale, the figures given on page 357 are reproduced in the table at foot of next page. As to the accuracy of these figures it should be noted that they have been obtained from a graph and are sufficiently accurate

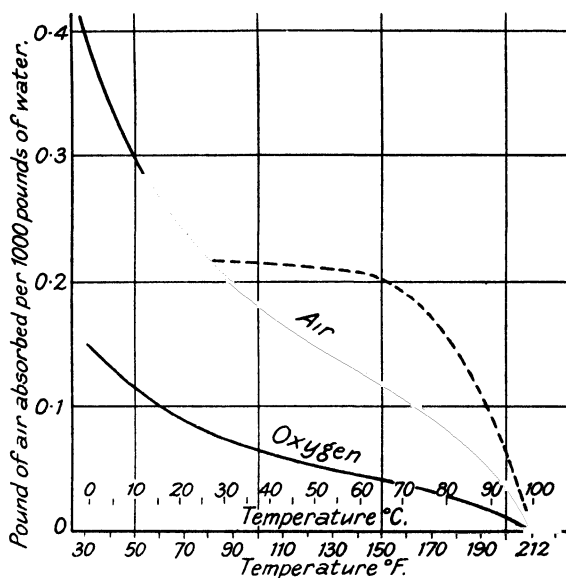


FIG. 219.

ABSORPTION OF AIR AND ATMOSPHERIC OXYGEN BY WATER.
(AIR PRESSURE PLUS VAPOUR PRESSURE 14.69 LBS./SQ. IN.)

or engineers if not for physicists. They are also graphed in Fig. 219.

IMPORTANCE OF CLOSED FEED SYSTEM. The following consideration will indicate the gain to be expected from the use of a closed feed system. Assume that the condensate leaves the condenser at 96° F. and that the partial air pressure is 0.3 in. or 1 per cent of the atmospheric pressure. The air contents are 0.001864 lbs. per 10,000 lbs. If this condensate is subsequently exposed to the atmosphere in the hot well or surge tank, the temperature being assumed to be unchanged, it will become saturated with air, taking up 0.1845 lb. of air. After passing through the system this additional air quantity will have to be handled by the extractor. If

the condensate were heated up to, say, 108° F. in or prior to the hot well, the total air contents would be reduced to 0.1696 lbs. and the extra amount to be handled by the extractor would be 0.1677 lb. With a closed feed system the air to be handled by the air extractor would, if the system be operating satisfactorily, be solely that due to leakage and to that contained in the make-up.

On page 347 there are details of a test in which, with a steam load of about 96,000 lbs. per hour, the calculated amount of air handled by the extractor was 9.35 lbs. This was with an open feed system, and the temperature of the condensate was 96° in the condenser and 108° after leaving a heater. Therefore the open feed system increased the air load by

$$0.1677 \times \frac{96,000}{10,000} = 1.61 \text{ lbs. per hour}$$

With a closed feed system, the air load would have been 7.74 lbs., so that in this particular case 83 per cent of the air was due to leakage and 17 per cent to an open feed system. This shows that there is more to be gained, as regards condenser operation, by careful attention to air leakage than by the adoption of closed feed systems. In this case the air load of 9.35 lbs. is the residue after very close

TABLE XXV

POUNDS OF ATMOSPHERIC AIR ABSORBED PER 10,000 LBS. OF WATER
(1 in. Pressure plus 1 in. vapour Pressure 14.69 lbs. per sq. in.)

Temperature	Oxygen	Nitrogen	Total
40° F	0.129	0.219	0.339
50° F	0.113	0.183	0.294
60° F	0.100	0.162	0.262
70° F	0.089	0.146	0.235
80° F	0.079	0.134	0.213
90° F	0.072	0.124	0.196
100° F	0.066	0.114	0.180
110° F	0.062	0.106	0.167
120° F	0.057	0.097	0.154
130° F	0.052	0.091	0.143
140° F	0.048	0.083	0.131
150° F	0.043	0.076	0.119
160° F	0.039	0.067	0.106
170° F	0.033	0.060	0.093
180° F	0.026	0.050	0.076
190° F	0.019	0.039	0.058
200° F	0.010	0.025	0.035
212° F	Nil	Nil	Nil

attention had been paid to all joints under vacuum, and it is still nearly six times as great as the air load due to the open aeration of the condensate.

Advantage of Closed Feed System from Boiler Standpoint. The greater importance attached to limiting air leakage as compared with a closed feed system is due to the question having been considered from a condenser point of view only. When judged from the boiler standpoint the closed feed system has greater value on account of the importance of eliminating oxygen to reduce oxidation or pitting. The reduction of the air contents in the condensate and feed system is therefore a matter of little moment as far as the effect on the condenser performance and air extractor capacity are concerned, whereas these factors are largely affected by the attention bestowed on the air tightness bestowed on glands and joints under vacuum. On the other hand, the reduction of air contents is of major importance when considered from the point of view of the boiler.

The air removed in the condenser, depending as it does upon the partial air pressure and the condensate temperature, varies with the condensate temperature for constant vacuum, assumed to be 28.5 ins., as follows—

Condensate Temperature.	Partial Air Pressure. Inches. Hg.	Air in Lbs. per 10,000 lbs. of Condensate.
91.67° F.	Nil	Nil
90.00° F.	0.076 in.	.00052
80.00° F.	0.466 in.	.00343
70.00° F.	0.759 in.	.00610
60.00° F.	0.977 in.	.00869

From the above figures (which ignore the effect on the condenser of high partial air pressures) it follows that the air contents are a minimum when—

- (a) the condensate temperature is a maximum, and
- (b) the partial air pressure is a minimum.

The first point depends upon the condenser design and the latter upon the air pump capacity.

Air Contents of Condensate. Few figures have been published regarding the air contents as determined by actual tests in power stations. Mr. P. L. Riviere* has stated that tests made at the Greenwich Power Station of the London County Council, showed that,

* *Proceedings Institution of Mechanical Engineers*, 1921 (Vol. I, p. 34).

condensate as discharged by the extraction pump at 70° F. contained 0.136 c.c. of oxygen per litre. Assuming that this gas volume is measured at 0° C. and 760 mm. the weight of oxygen present is 0.0001945 grammes, corresponding to 0.001945 lbs. per 10,000 lbs. Allowing for the accompanying nitrogen according to the figures given on page 359 for 70° F., the total weight of air per 10,000 lbs. of condensate is 0.00513 lbs. From the figures on the same page, the amount required to saturate water exposed to the atmosphere at this temperature is 0.235 lbs., so that the water at this point had a relative aeration of 2.18 per cent. If the volume of oxygen 0.136 c.c. were measured at the water temperature, the relative aeration would be reduced to 1.89 per cent.

Mr. J. E. Lea* has stated that the relative oxygenation of condensate drawn at 80° F. from plant fitted with Edwards air pumps was 57.5 per cent, and in condensate at 110° F. from plant fitted with rotary air and extraction pumps, 53 per cent. Allowing for the nitrogen, these figures correspond to an air contents per 10,000 lbs. of condensate of 0.1225 and 0.0885 lb. respectively.

When air saturated water, i.e. water with a relative aeration of 100 per cent is heated, the released air tends to separate and rise to the surface as bubbles. This tendency of the air is retarded by the surface tension and by the viscosity of the water. Both these properties of water decrease with increasing temperature and eventually at approximately 160° F. the bubbles commence to rise. Until they are liberated the water has a relative aeration in excess of 100 per cent or it is supersaturated. The air contents, with rising temperature, are shown approximately by the dotted line in Fig. 219. If water be agitated when it is heated, the retarding effect of the surface tension, etc., is overcome, and the relative aeration remains at about 100 per cent.

A similar liberation of contained air is caused when water at a certain temperature is introduced into a vacuum space of which the temperature is about 20° to 25° below that of the incoming water.

The heat corresponding to excess temperature of the entering water is absorbed by the evaporation of part of the water; so that thermal equilibrium is reached. In the rapid evaporation of the water resulting from this temperature difference, the steam forms round the air bubbles as nuclei and in leaving the water carries the air with it.

Details of De-aerating Plant. Such special apparatus is termed a de-aerator, but many closed feed systems use the main condenser as a de-aerator by passing all the make-up feed water through it.

* *Proceedings Incorporated Municipal Electrical Association, 1920.*

An arrangement of this latter type, due to Messrs. Hick, Hargreaves & Co., Ltd., is illustrated diagrammatically in Fig. 220. The condensate passes from the condenser *A* by the extraction pump *B*, to the feed tank *D*. This tank is closed to the atmosphere and a steam pipe allows steam to enter in such amount that there is always a slight escape at a vent pipe. From the feed tank *D*, condensate is drawn by the lift pump *E*, and discharged through a surface heater *X* to an overhead tank *Z*, which is also steam sealed. From this tank the boiler feed pump draws. Make-up water enters the lower open tank *H*, the flow being controlled by a float in tank *D*, so that a fall in the water level in *D* opens the

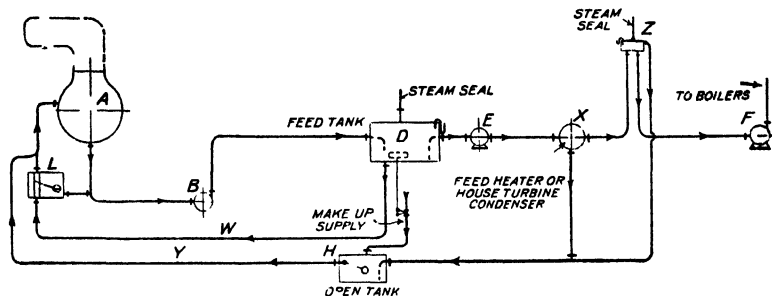


FIG. 220

DE-AERATING SYSTEM

(Hick Hargreaves & Co., Ltd.)

make-up supply. As the level in *H* rises, a float opens a valve leading to pipe *Y*, allowing the water to pass to the condenser and so into the system after being de-aerated in the condenser. To ensure stable working of the extraction pump at low loads, there is, in the closed tank *L*, a float controlled by the water level in the pipe between condenser and extraction pump. As this level falls, a valve is opened admitting water from tank *D* to the condenser. In order to conserve the condensate and any contained heat, the water of condensation from the heater *X*, and any overflow from the tank *Z*, are led into the open tank *H*, and thence returned to the condenser.

The same system but arranged to include an evaporator for introducing the make-up feed water and a house service turbine is shown in Fig. 221. In this the main path of the condensate will be seen to be condenser *A*, extraction pump *B*, water meter *O*, sealed feed tank *D*, lift pump *E*, house service condenser *X*, overhead sealed tank *Z*, feed pump *F*, and then to boilers. The overflow from tank *Z* and the condensate from condenser *X* pass to the open tank

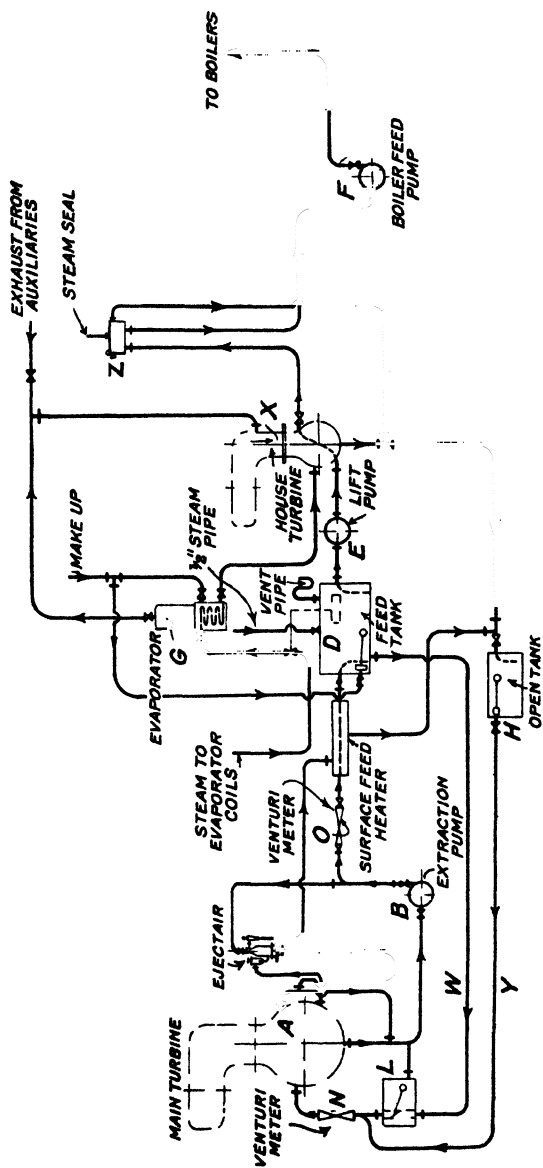


FIG 221

DE-AERATING SYSTEM INCLUDING EVAPORATOR

(Hick, Hargreaves & Co Ltd)

H, and so to the main condenser with the make-up. The make-up water enters the evaporator, and the produced steam passes as shown to the house service condenser *X*, where it gives up its heat to the condensate, and then passes down to the tank *H*. Steam from the evaporator coil also enters the condenser *X*, and mingles there with the produced steam. The amount of make-up evaporated is controlled by a float in tank *D* which regulates the amount of steam passing to the evaporator coil. This arrangement also shows the two-stage steam-jet air extractor with the water from the inter-condenser going back to the main condenser or to the extraction pump suction. The steam and air from the second stage pass through a small surface heater whence the air escapes and the water goes on to the open tank *H*. The arrangement for ensuring that the extraction pump *B* works satisfactorily at low loads is the same as in the previous scheme. It will be noticed that two Venturi meters are shown. The meter between the extraction pump *B* and the surface feed heater will register more than the amount of condensate corresponding to the steam load on the condenser, if condensate is returned from tank *D* to the condenser *A* by tank *L*. A second Venturi meter is therefore fitted at *N*, the difference between the readings of the two meters giving the steam load on the plant.

An independent de-aerating plant by Messrs. Hicks, Hargreaves & Co., Ltd., is shown in Fig. 222. It consists mainly of a de-aerating chamber *A*, air ejector *B*, and heater *C*. When used in connection with an open feed system, the feed water enters the heater from the hot well being aspirated by the partial vacuum maintained in the de-aerating chamber *A* by the ejector *B*. The feed passes through the heater where it condenses the steam from the ejector, and any other steam that may be available or needed. The heated water enters the chamber through nozzles which divide it into a fine spray. Being at a temperature above that corresponding to the vacuum in the chamber *A*, the water flashes and the steam thus formed carries the air away. The air is removed by the ejector. The water of condensation from the heater drains to the tank *D*, when it is drawn through a float controlled vacuum trap to the chamber *A*. The flow of water to the apparatus is controlled by a float-controlled butterfly valve which maintains a constant head on the extraction pump.

 **The Contraflo Closed Feed System.** The closed feed system which has been developed, mainly for marine use, by the Contraflo Condenser Co., Ltd., is shown diagrammatically in Fig. 223. In this the main path of the condensate is from the condenser bottom to the extraction pump (a vertical pump is shown) and on to the

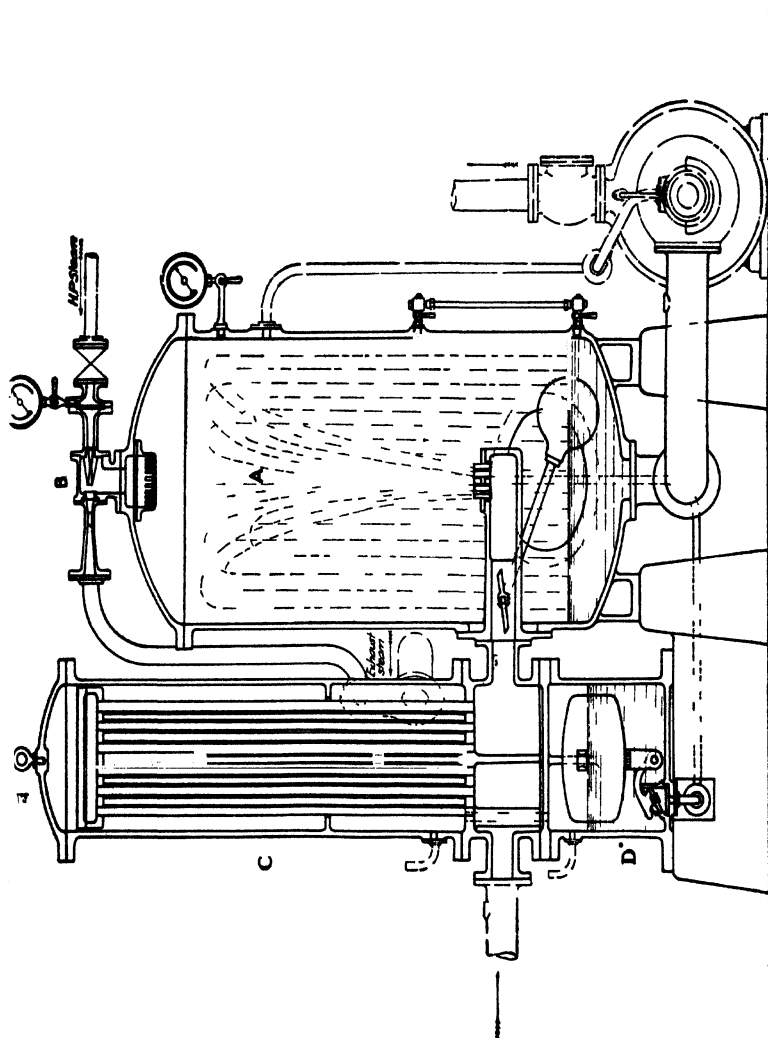


FIG 222
ARRANGEMENT OF HICK-HARGREAVES INDEPENDENT DE-AERATING PLANT.

auxiliary heater D, the boiler feed pump and the main surface feed heater, and so to the boiler. A two-stage steam-jet air ejector is shown with inter-condenser. This inter-condenser or receiver is supplied with condensate by a shunt on the extraction pump discharge. This condensate is returned to the condenser by the usual loop pipe. The discharge from the second stage is passed into the auxiliary heater D, whence the air escapes to the atmosphere. Between the auxiliary heater and the feed pump, a relief valve is provided so that surplus water, indicated by an increased pressure in this pipe, passes to the feed tank. The make-up feed water is drawn into the condenser, through the pipe F, from the feed tank, the flow being regulated by the valve shown on this pipe. If too much water passes into the condenser the surplus is returned to the tank through the relief valve. The pipe F draws from the feed tank through two connections as shown. If the level in the tank falls too low the valve H is closed and valve J opened by the float. This reduces the amount of water drawn from the feed tank and allows make-up water to be drawn from the reserve tank. Thus, with this system as with the one described earlier, the de-aeration is done in the condenser. To the main surface heater there is led auxiliary exhaust steam which heats the condensate. The water of condensation passes through the auxiliary heater where it gives up heat, and then to the feed tank. A control valve and a throttle valve are provided before and after the feed pump to maintain a constant pressure on the extraction pump discharge, by throttling the feed pump discharge, should the extraction pump discharge pressure fail.

Whilst the arrangement shown in Fig. 223 is primarily laid out for a marine installation, its adaptation for a land plant is obvious. The usual source of make-up water would replace the reserve feed tank and the oil separators and oil filter, throttle and control valves would not be required.

The special valves used in this arrangement are shown in fuller detail in Fig. 224. The by-pass valve is connected to the main turbine casing and is closed when the pressure at the point of connection is above a certain value. On lower loads the pressure falls and the valve opens, allowing water from the extraction pump discharge to pass back to the condenser. If this is in use the valve in the vertical pipe F, just above the feed tank in Fig. 223, can be closed.

The throttle and control valves maintain a constant pressure at the feed-pump suction. Water leaks back through a small hole in the piston on the lower end of the throttle valve, which is of the

non-return type, to the feed-tank. This leak-off is closed by the control valve when the pressure is normal, but it is restarted as soon as the pressure falls. This flow of water lowers the pressure under the throttle valve which therefore partially closes.

■ **CLOSED FEED SYSTEM.** A closed feed system with the Kinetic rotary pump is illustrated in Fig. 179, which has been briefly described on page 291. The condensate is passed through the inter-condenser to the boiler feed pump suction, surge capacity

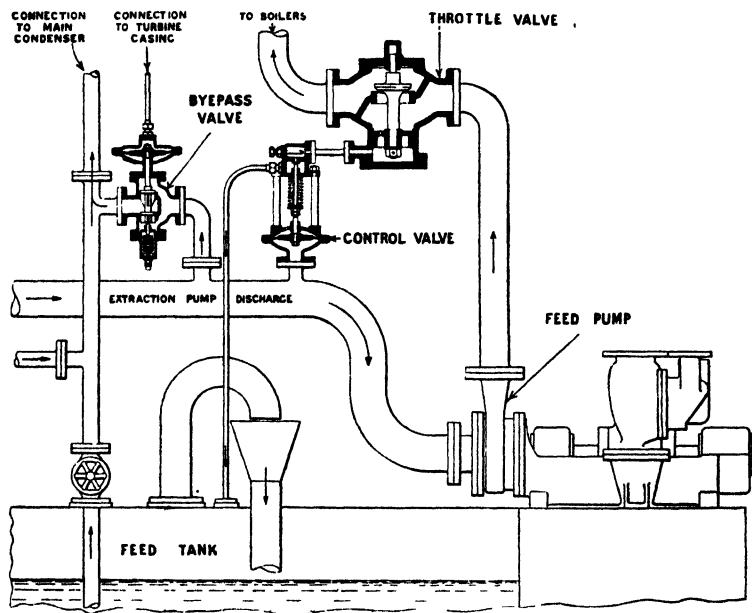


FIG. 224

ARRANGEMENT OF VALVES IN CLOSED FEED SYSTEM.

(Contraflo Condenser Co.)

being provided in the overhead tank. Reliance is placed upon the column of water above the inlet from the heater to prevent air entering the comparatively air-free condensate. To supply the make-up water, there must be a small downward motion in the vertical pipe, hence some air enters the system at this point.

CONTRAFLO DE-AERATING SYSTEMS. A closed feed system as supplied to land plant devised by the Contraflo Condenser Co., Ltd., is shown diagrammatically in Fig. 225. In this scheme the condensate is de-aerated in a separate chamber under a partial vacuum and

the air returned to the condenser, where it is handled by the air ejector in the usual way. The condensate extraction pump passes the condensate through a heater where its temperature is raised to about 170° , and then into the de-aerator. In this a vacuum of about 22 ins. is maintained by a restricted connection to the condenser. The condensate temperature is about 20° above that corresponding to the 22 in. vacuum, and to absorb the excess heat some water is evaporated suddenly. This violent ebullition drives off the air. The air is drawn off to the condenser with the vapour which has been formed. Make-up feed water is produced by a double-effect evaporator, the steam supply to which can be controlled by the water level in the de-aerator. The produced

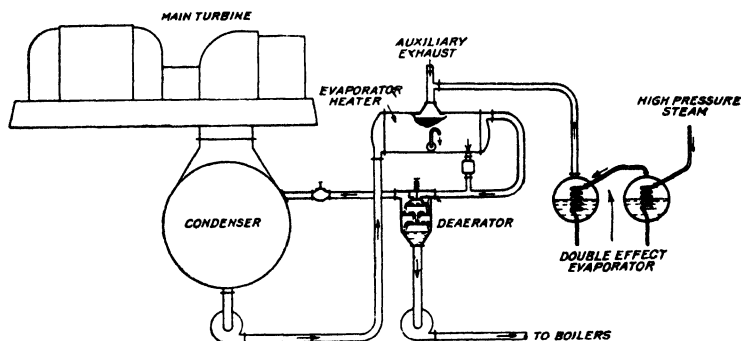


FIG. 225.

CLOSED FEED SYSTEM AS APPLIED TO LAND PLANT.

(Contraflo Condenser Co.)

steam is led to the heater, and, in condensing, heats the main condensate, after which it passes to the de-aerator with the main condensate.

A system, also due to the Contraflo Co., somewhat similar to that in Fig. 225, is shown in Fig. 226. Here two or more main generating units are working in connection with one house service turbine and one de-aerating plant. The extraction pumps pass the condensate into the de-aerator. The ejected air with its contained vapour joins the house service turbine exhaust and passes through the tubes of the de-aerator. Any steam not condensed here passes to the evaporator heater, and joining with the steam produced in the evaporator is finally condensed. The de-aerated water is drawn by a lift pump and discharged through the heater to the boiler feed pump. This diagrammatic sketch does not show the relief valve before the boiler feed pump nor the float control on the

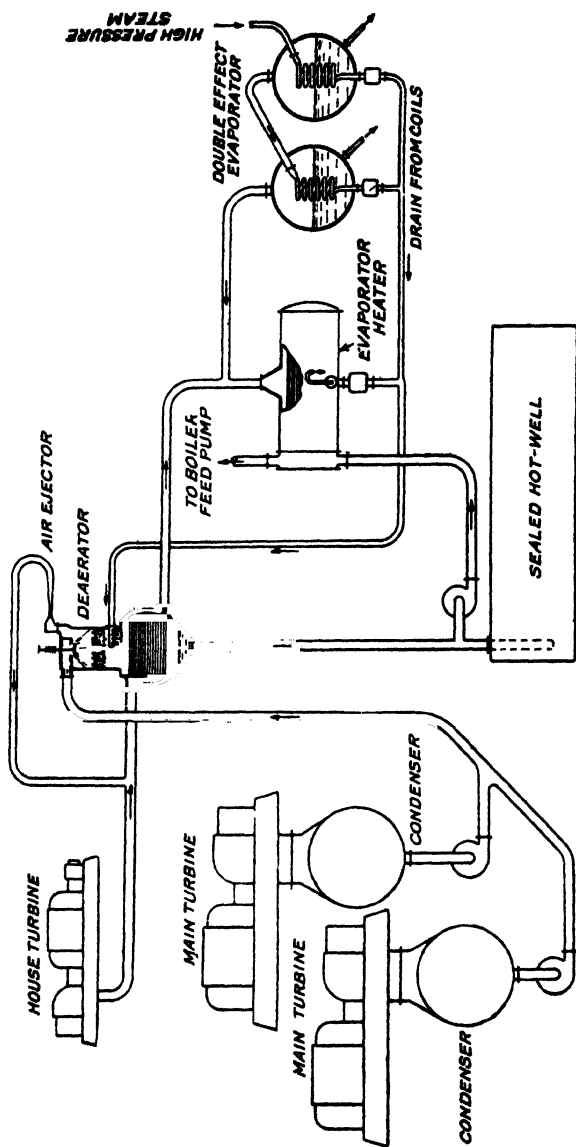


FIG 226.
CLOSED FEED SYSTEM
(Contraflo Condenser Co.)

steam supply to the evaporator, but these arrangements are easily made.

It should be noticed that there is no thermal loss with this method beyond the inevitable radiation loss. The steam produced by contact

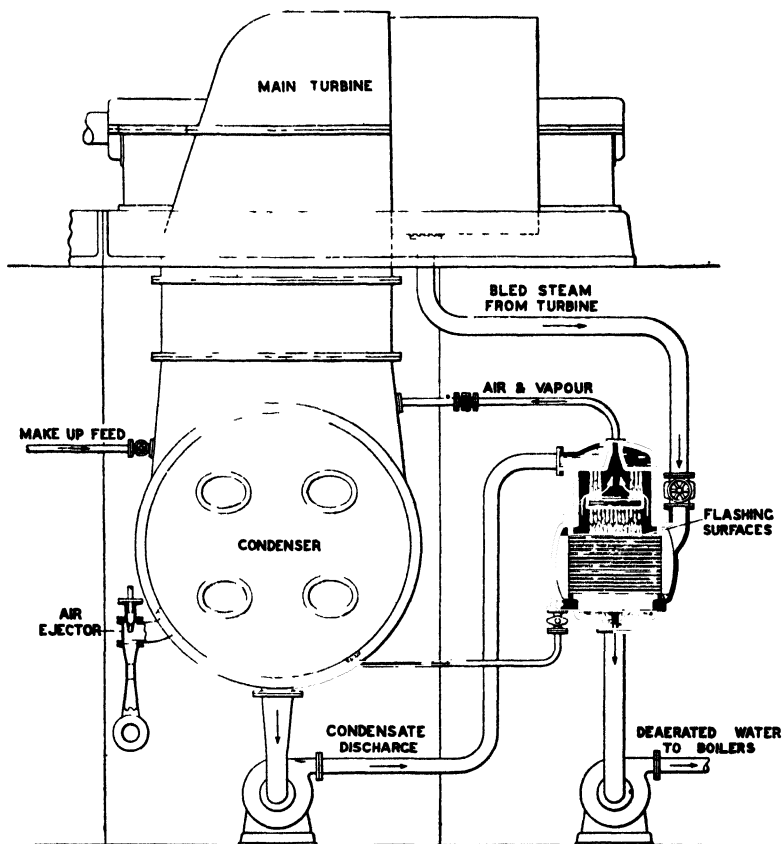


FIG. 227.

ANOTHER ARRANGEMENT OF DE-AERATING SYSTEM.

(Contraflo Condenser Co.)

with the tubes rises and is condensed by the water entering above, so that all the heat in the steam used in the de-aerator is ultimately transferred to the condensate. A certain amount of heat is in constant circulation between the hot tubes, the water in contact therewith, the steam formed and the water met by the ascending

steam, but the thermal losses are limited to radiation and leakage if any.

Another "Contraflo" arrangement of de-aerating plant is shown in Fig. 227. Here the make-up feed is admitted directly to the

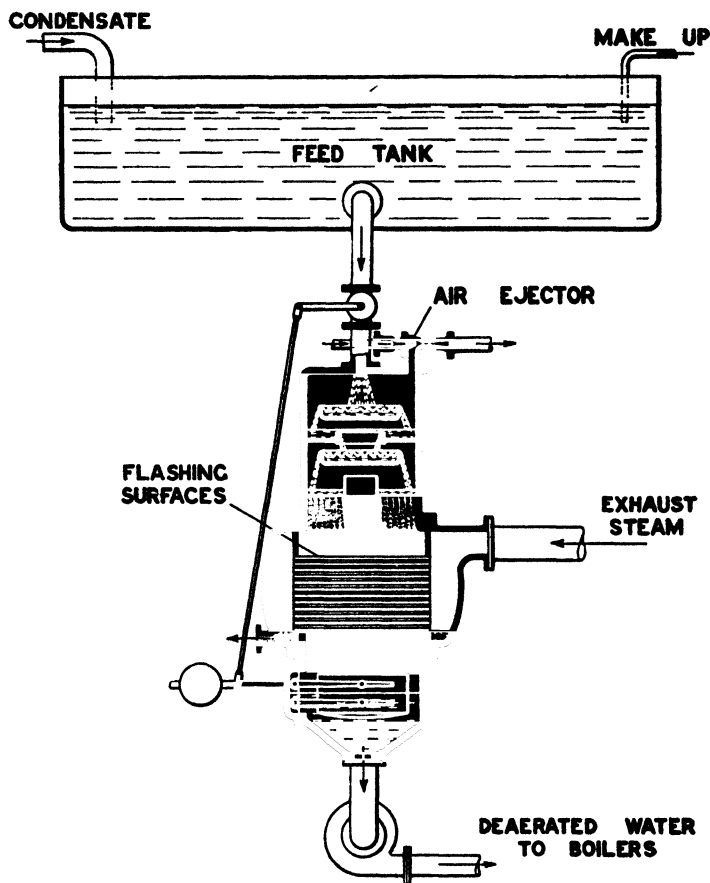


FIG 228.

DE-AERATOR SEPARATE FROM CONDENSING PLANT
(Contraflo Condenser Co.)

condenser, the control not being shown. The condensate is passed into the de-aerator, where it is sprayed on to the surface of tubes through which steam abstracted or bled from the main turbine is passing. This bled steam is at such a pressure that it heats the water suddenly, and evaporating some, causes the air to be driven

out of the water. The air then passes to the main condenser. This arrangement ensures approximately constant temperature in the de-aerator as if the load on the main turbine is reduced, then the quantity of steam abstracted and the amount of condensate are both reduced proportionately.

If this type of apparatus is used as a de-aerator apart from a condensing plant, it could be arranged as shown in Fig. 228. Here it is fitted with a separate ejector to maintain the necessary vacuum. In Fig. 229 the plant is shown diagrammatically as arranged for a test. The results of the test are given on page 374, the various pressures and temperatures being measured at the points indicated in Fig. 229.

The oxygen contents were measured by the iodine precipitation which will be described later. The results show that the initial aeration of the water does not affect the final aeration.

The Mirrlees-Watson Closed Feed System. A closed feed system introduced by the Mirrlees-Watson Co., Ltd., of Glasgow, is shown in Fig. 230. Examination of this figure will show that the condensate passes from the condenser through the following pieces of apparatus in turn:—Control box *B*, condensate pump *C*, inter-condenser *D* of the ejector air pump *J*, pressure tank *E*, Venturi meter *H*, drain heater *P*, ejector heater *H*, and bled steam heater *O*, and then by the feed pump to the boiler. By-passes are arranged for the three heaters *P*, *H*, and *O*. To ensure stability of the condensate pump at low loads, condensate is passed back from the pressure tank *E* to the control box, the flow being controlled by a float in the control tank *B*. This arrangement involves the use of two vacuum traps *N* and *Q*.

When this arrangement is modified to include an evaporator, steam-driven pumps, jet inter-condenser for the air ejector, it becomes as shown diagrammatically in Fig. 231. The main path of the condensate remains as in Fig. 230, and from the figure there will be seen the discharge from the jet inter-condenser through a loop pipe to the condensate pump suction, and the exhausts from the turbines driving this pump and the feed pump to the main heater, where they join the steam bled from the main turbine. The make-up water may be taken from the circulating water discharge or from an external supply, and passes through a heater *R*, where it is heated by drains *en route* to the control box *B* to the evaporator *N* by feed regulator *P*. The steam generated passes to the main heater, and the water formed in this heater goes via the vacuum trap *M* to the control box *B*, where it enters the condensate system.

It will be noticed that the main heater *K* is also marked de-aerator.

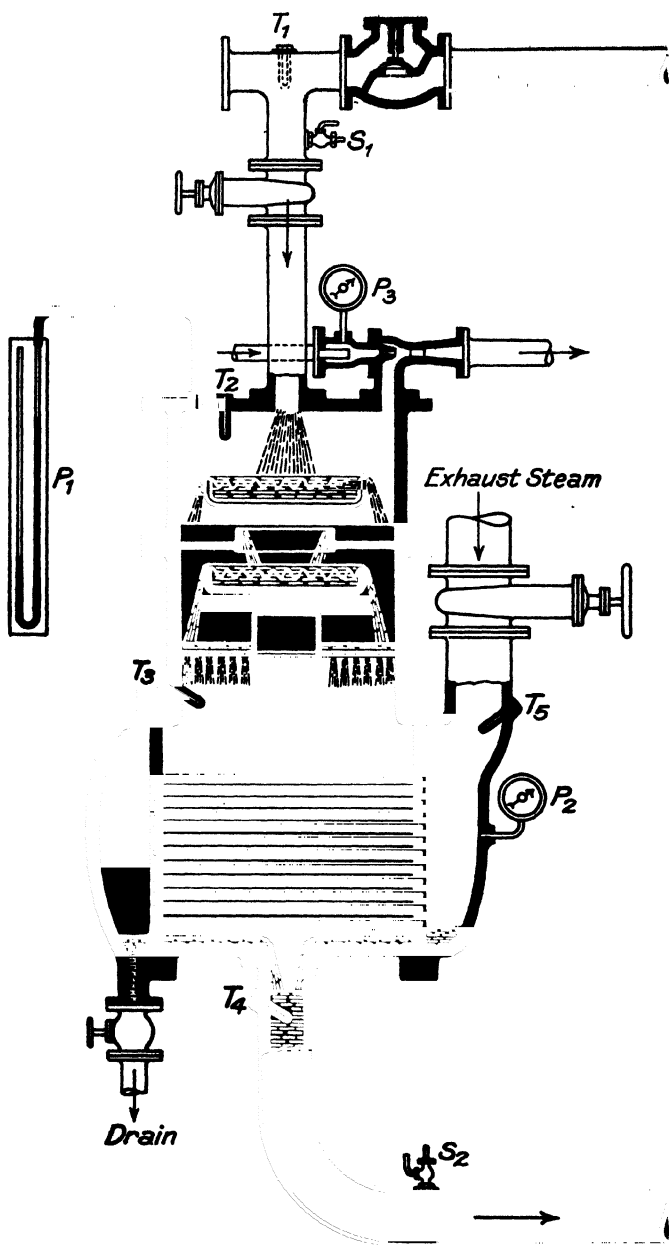


FIG. 229.
DE-AERATOR TESTING PLANT.
(Contraflo Condenser Co.)

The arrangement of the de-aerator is shown diagrammatically in Fig. 232. This apparatus works on the same principles as other de-aerators, i.e. by introducing heated water into a space at a

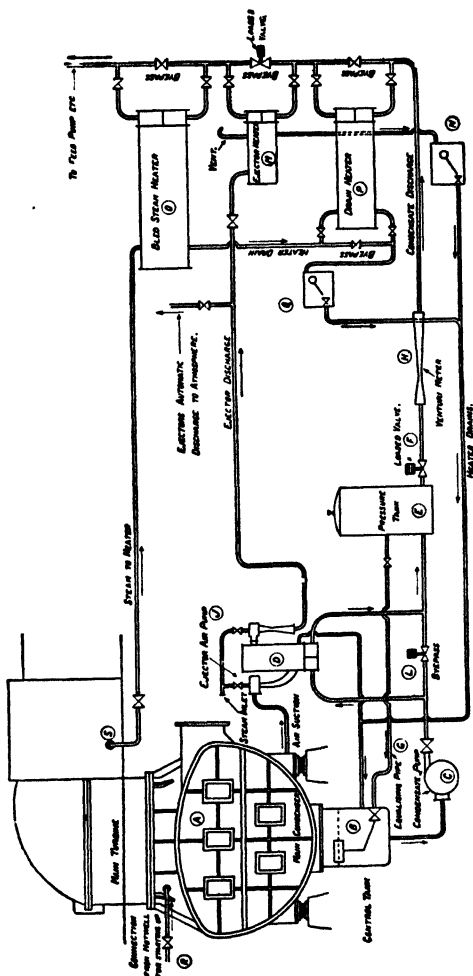


FIG. 230.

THE MIRREES-WATSON CLOSED FEED SYSTEM.

pressure below that corresponding to the temperature of the water. In Fig. 232 the main parts are A the de-aerator proper, B a feed water heater, C a vapour condenser, D an air ejector, and E the ejector steam condenser. Feed water travels in turn through E, C, and B, and is raised to the required temperature. The heating

Another closed feed system by Messrs. Mirrlees-Watson, Ltd., is illustrated in Fig. 232. The condensate passes from the condenser *A* to the collecting tank *G*, whence it is drawn by the extraction pump *B*. This pump delivers it through sluice and

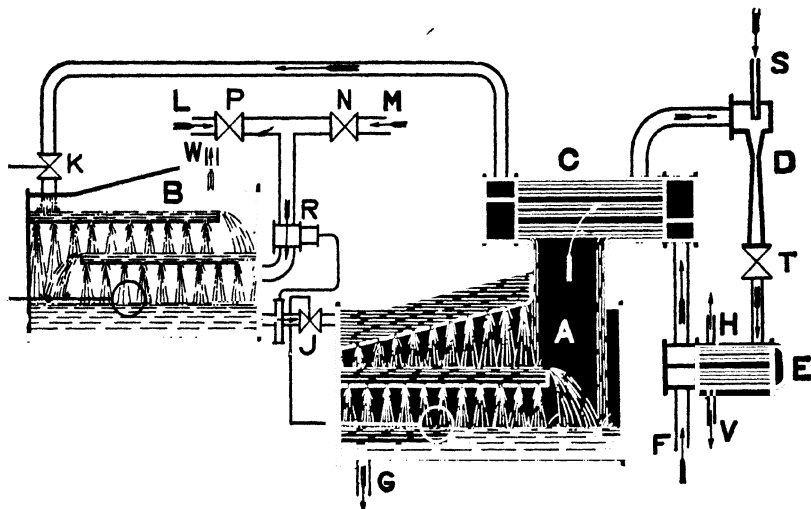


FIG. 232.

DE-AERATOR.

(Mirrlees Watson Co., Ltd.)

non-return valves to an overhead tank *D*, which may be a measuring tank such as a Lea Recorder. This tank is steam sealed to preserve the closed feed system. The extraction pump *B*, is maintained in stable working condition by the connection *K* between the tanks *D* and *G*. If the water level in *G* falls, a float-controlled valve opens the connection *K* and water passes back from *D* to *G*, thus restoring the level in the latter tank. The water passes over the weir *F* at the rate at which it enters the tank *G* from the condenser, the flow over the weir not being affected by the passing back of a certain amount of water. The other items in the plant are self-explanatory. The inter-condenser in the ejector *C* is supplied with a constant quantity of water from the overhead tank *D*. The steam from the second stage proceeds to a heater *E* through which condensate passes from the overhead tank *D* to the feed pump *J*. The resulting condensate, formed in heater *E*, passes to tank *H* and is admitted to the condenser by a float controlled vacuum trap. If, on light loads, the ejector steam heats the condensate too much, this hot water can be passed to the

condenser by the connection *M*, operated either by hand or automatically by a thermostatic valve.

Instead of a weir recorder as shown in Fig. 233, a closed tank and

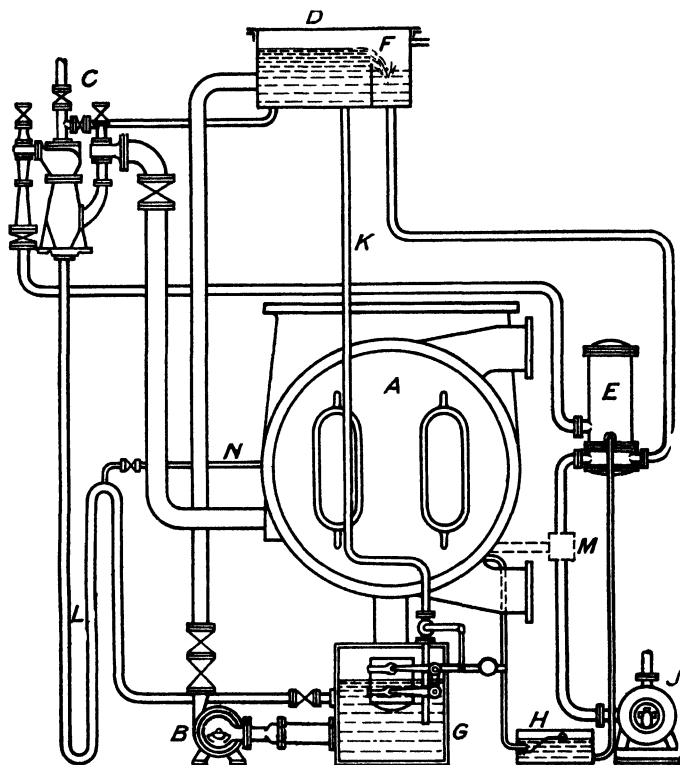


FIG. 233.

MIRRLEES-WATSON CLOSED FEED SYSTEM.

venturi meter can be used, the arrangement being indicated in Fig. 234.

Elliott De-aerating System. The same principle of introducing heated water into a vacuum chamber has been developed very largely in the United States by the Elliott Co., of Pittsburg. They generally use separate chambers for heating and air separation, and in Fig. 235 there is shown diagrammatically their usual arrangement. The cold raw water enters the condenser shown at the top of the figure. From this it passes into the heater through a float controlled valve maintaining a constant level in the heater. Exhaust

steam is led into this heater and is condensed in the water, submerged nozzles being usually used. The water then enters the bottom vessel, the separator, also through a float controlled valve. A

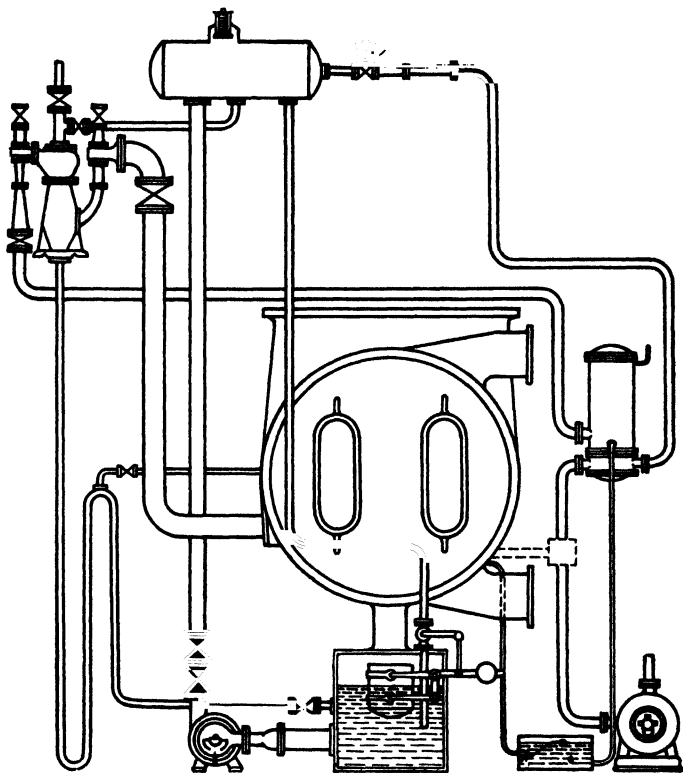


FIG 234.

ARRANGEMENT INCLUDING CLOSED TANK AND VENTURI METER
(Mirrlees-Watson Co., Ltd.)

vacuum is maintained in the separator by a single-stage steam ejector and the air is removed by this ejector. The treated water then passes away to the feed pump and so on to the boilers. The ejector discharge is led to the condenser previously mentioned where the steam is condensed, giving up its heat to the in-coming cold water. The condensate formed thereon flows by gravity back to the heater.

In principle this arrangement will be seen to be similar to the apparatus illustrated in Fig. 228.

The details can be varied to suit any special circumstances. Thus if it be desired to reduce the ejector steam consumption, the condenser can be divided into two units, one portion being placed

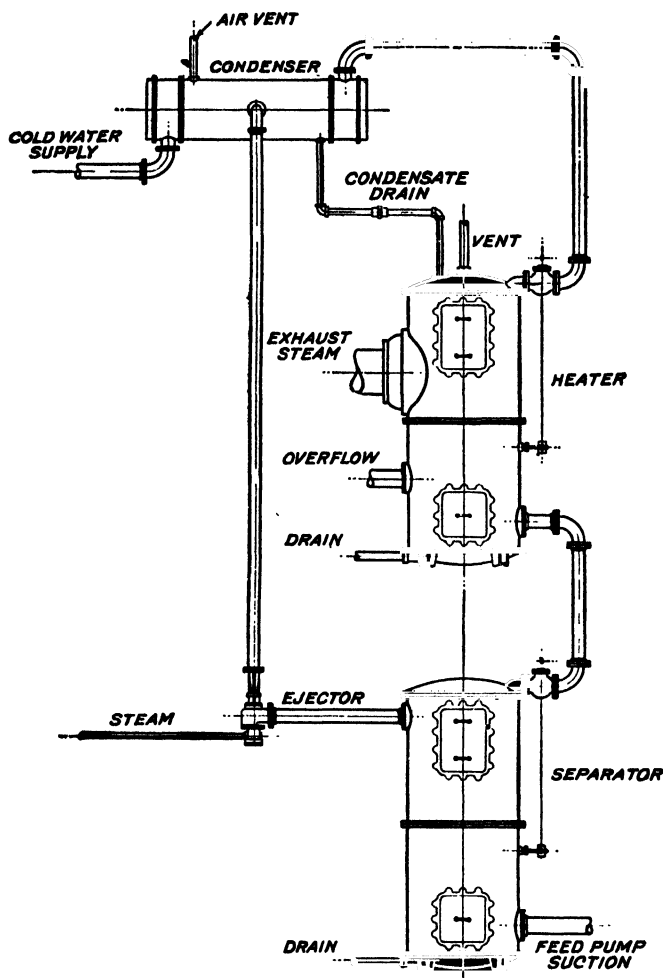


FIG. 235.

ELLIOTT DE-AERATING SYSTEM.

prior to the ejector. Alternatively a rotary or a reciprocating air pump could be used instead of the ejector. In some cases with an ejector a contact heater is used instead of the surface condenser.

Details of a similar arrangement are shown in Fig. 236. In this

the water enters the condenser by the flange *C*, and passes through to the heater entering by the flange *H*. A by-pass allows some of this water to go to the contact heater and condense the ejector

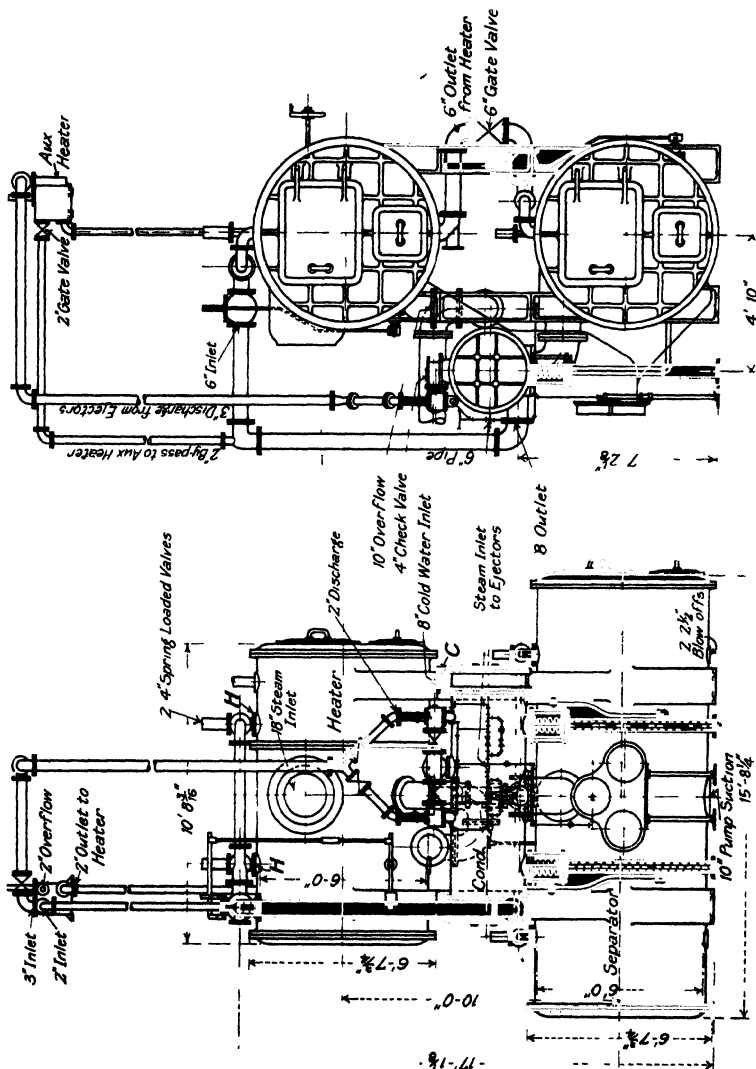


FIG 236.
ELLIOTT DE-AERATING PLANT

steam. This plant is capable of treating 225,000 lbs. of water per hour, and the overall dimensions are approximately 15 ft. 8 ins. $\times$ 11 ft. 6 in. $\times$ 18 ft. high.

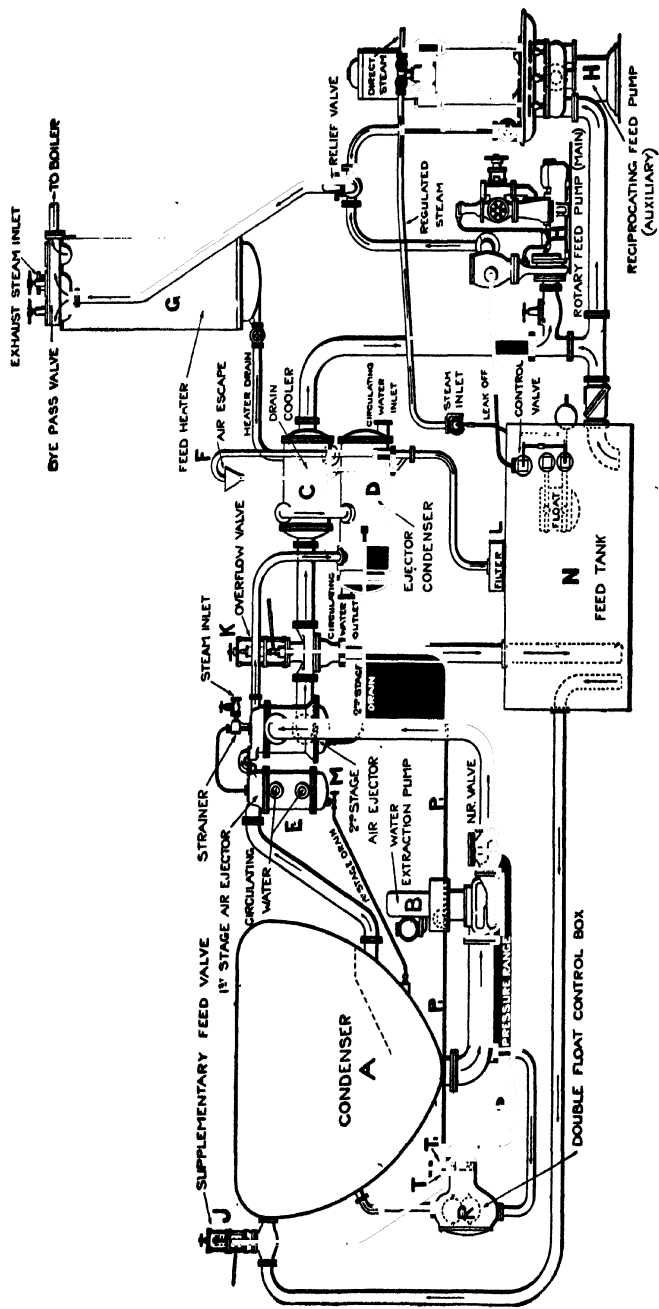


FIG. 237.—THE WEIR CLOSED CIRCUIT FEED SYSTEM.

Diagrammatic Arrangement for Merchant Service Vessels.

- | | | | | | |
|---|------------------------------|---|---|----------------|---|
| A | Condenser | H | Main (Rotary) and Aux. (Reciprocating) Feed Pumps | P | Pressure Pipe to Float Control Box and Supplementary Feed Valve |
| B | Rotary Water-extraction Pump | I | Supplementary Feed Valve | P ₁ | Pressure Pipe to Overflow Valve |
| C | Drain Cooler | K | Overflow Valve | R | Double Float Control Box |
| D | Ejector | L | Filter for Heater Drain | T | Relay Valve (Supplementary Feed Control) |
| E | 1st Stage Air Ejector | M | Air Ejector 1st Stage Drain to Condenser | T ₁ | Relay Valve (Overflow Control) |
| F | 2nd Stage Air Ejector | N | Feed Tank | | |
| G | Feed Heater | | | | |

Weir Closed Feed System. Another closed feed system arranged by Messrs. G. and J. Weir is shown diagrammatically in Fig. 237. In this the condensate passes from the condenser A in

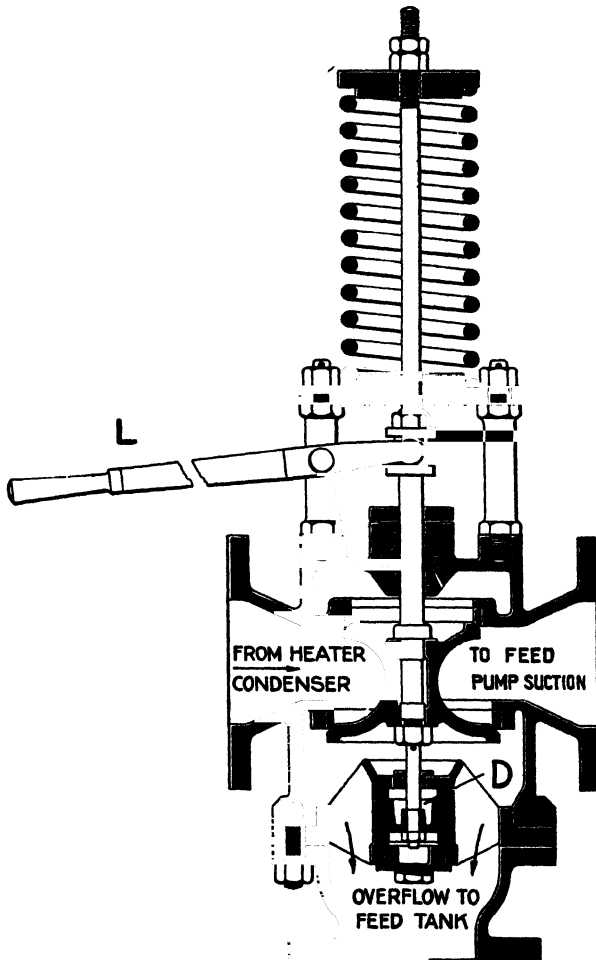


FIG 238
DETAILS OF WEIR OVERFLOW VALVE

turn through the extraction pump B, the after-condenser, drain cooler C, to feed pump H, and thence through the main feed heater G, to the boilers. The air ejector is of the two-stage type with surface inter- and after-condensers, the second stage discharging

to the ejector condenser D, through which circulating water is passed. When starting up D condenses the second stage steam, but normally this steam is condensed by condensate passing through the after-condenser. In the illustration there will be noticed a double float box R. If the condensate accumulates in the condenser the upper float rises and closes a leak-off valve T¹, thus raising the pressure in

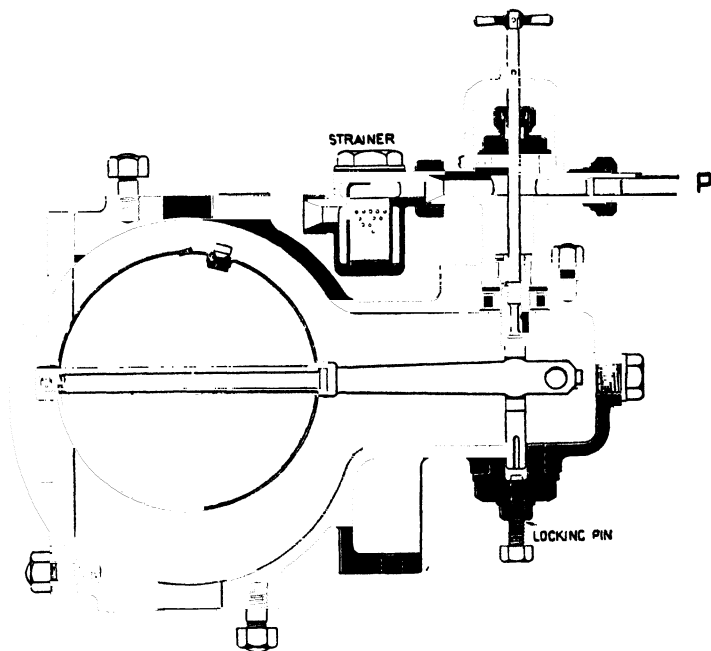


FIG. 239.

SECTION OF WEIR FLOAT CONTROL BOX.

pipe P¹. This, in turn, opens the overflow valve K, and releases the surplus water to the feed tank N. Similarly if the water level falls at the extraction pump suction the low float opens the leak-off valve T, and so opens the supplementary feed valve J, admitting feed from tank N to the condenser.

For its successful operation this closed feed system requires two special valves, the overflow or relief valve and the supplementary feed valve, and the float control box.

The overflow valve is shown in detail in Fig. 238. Any pressure on the extraction pump discharge in excess of that for which the valve is set, opens the double-beat valve and condensate passes

down to the tank. The dash-pot D prevents chattering, and the lever L allows the valve to be hand-operated to ascertain that it is moving freely.

In Fig. 239 there is a section of the float control box connected to the extraction pump suction, but only showing the low level

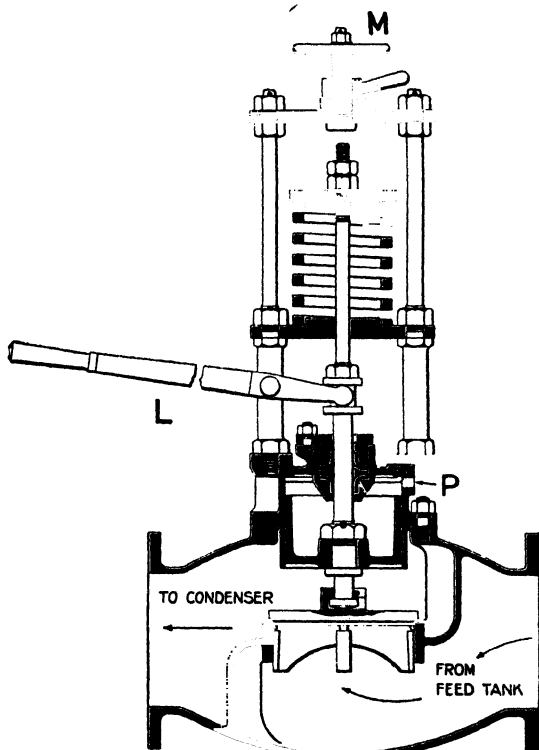


FIG. 240.
SUPPLEMENTARY FEED VALVE.
(G. & J. Weir, Ltd.)

float. With the float at its normal level the spindle on the right is held up so that the valve shown is on its seat. Hence the pressure from the extraction pump discharge brought by the pipe P, is transmitted undiminished through the strainer shown to the supplementary feed valve shown in Fig. 237. Any fall of the float opens the valve and releases the pressure.

The effect of this pressure release is seen in Fig. 240. This control valve is normally closed by the pressure from connection P against

the spring shown. Thus the valve opens when the water level in the extraction pump suction falls and extra condensate is passed into the condenser.

The float control valve used in this latter system is similar in its function to that used in the system illustrated in Fig. 221.

Brown-Boveri Feed Systems. Brown-Boveri Co. have developed a method of dealing with the condensate, which although

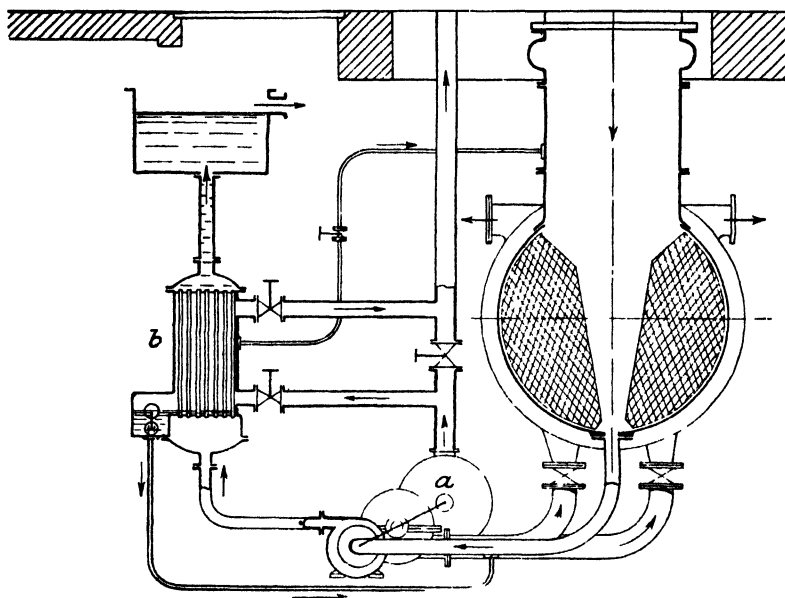


FIG. 241.

BROWN-BOVERI ARRANGEMENT OF AUXILIARIES.

not exactly a closed feed system is so closely related thereto, that it will be examined here. By a slight addition not shown in the illustrations of this method, it would be made into a closed feed system.

When utilizing the exhaust steam from turbine-driven auxiliaries, trouble has always been caused by the exhaust steam being in excess of requirements at low loads. This has led many engineers to use motor driven circulating pumps, but Brown-Boveri have introduced the arrangement shown in Fig. 241. Here on one axis will be seen the condensate extraction pump, the circulating pump, and the driving turbine A. The condenser is of the O.V. type described on pages 97 to 101. The turbine exhausts into a surface heater

B, and any steam not condensed there passes out to the main turbine in which it expands to the condenser pressure doing useful work. The pressure of the auxiliary exhaust is determined by the pressure at the main turbine connection, so that at high loads the exhaust pressure and therefore the steam temperature in the heater, is a maximum. The condensate temperature remaining approximately

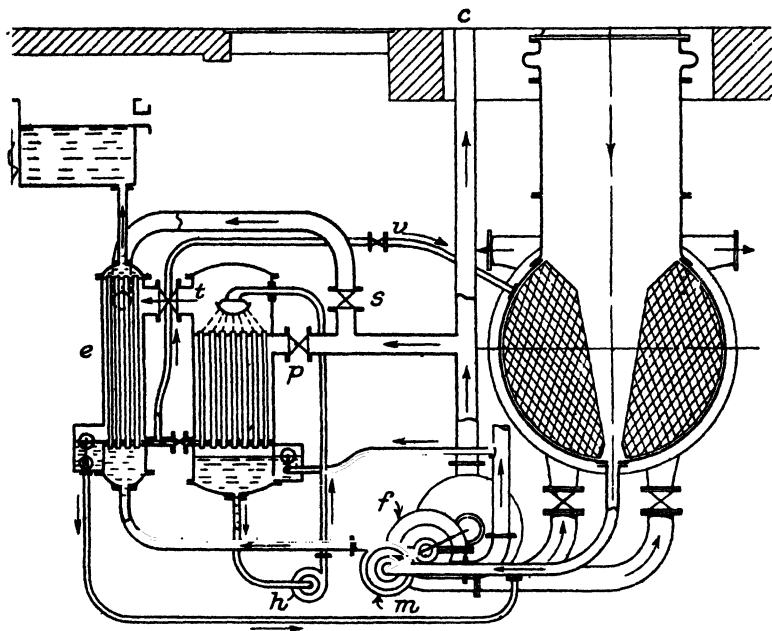


FIG. 242

BROWN-BOVERI ARRANGEMENT INCLUDING EVAPORATOR.

constant, the heat transferred to the condensate varies with the steam temperature. This, however, varies with the load on the main turbine, and as the condensate also varies with the load, the discharge temperature of the condensate is approximately constant at all loads. Should the heater require more steam than the auxiliary turbine can exhaust, steam can be taken from the main turbine to assist in the heater. As the auxiliary turbine may have to exhaust into a vacuum at low loads, the steam space of the heater is connected to the main condenser to facilitate the action.

When an evaporator is added to this system it becomes as illustrated in Fig. 242, in which the evaporator is interspersed between the auxiliary turbine and the heater. The auxiliary turbine exhaust

passes into the evaporator, any surplus going on to the main turbine at C, or if deficient, being supplemented by steam from the main turbine. Water from the circulating pump discharge F enters the base of the evaporator through a float-controlled valve. This water is kept circulating through the evaporator by the pump *h*, and the produced steam passes to the heater *e*, through which the condensate is circulated by the extraction pump *m*. The water of condensation formed in both the evaporator and heater returns through a float-controlled valve to the extraction pump suction.

The evaporator can be isolated by closing valves *p* and *t*, and steam admitted direct to the heater by opening valve *s*. The heater also can be isolated by closing valves *s* and *p*, and the auxiliary turbine can exhaust to the main turbine.

To provide for the possibility of the auxiliary turbine exhausting into a vacuum, both the heater and evaporator are connected to the condenser by pipe *v*.

Owing to the control upon the auxiliary exhaust exercised by the pressure at point *c* in the main turbine, the temperature difference in the evaporator, and therefore the amount of vapour produced therein, varies with the load. If the auxiliary exhaust is more than is required to produce the necessary make-up at any load, it can be adjusted by partially opening valve *s* and allowing some steam to go directly to the heater.

To make this system into a closed feed system, the discharge from the heater *e* would be led to the boiler feed pump suction with a relief valve, allowing surplus water to escape to a sealed feed tank. The make-up water would then be taken from this tank and not directly from the circulating water discharge.

Closed Weir Type Condensate Meter. The use of open-weir type condensate meters has been condemned in the past as tending to aerate the feed water. To overcome this objection the Lea Recorder Co. has introduced an arrangement in which the weir is in a closed vessel, connected directly to the condenser and therefore under vacuum. This is illustrated in Fig. 243.

Method of Ascertaining Oxygen Contents. The method usually employed in ascertaining the oxygen contained in water is that due to Winkler, in which the oxygen is replaced by iodine, the latter being measurable. A sample of the water to be tested is obtained in a stoppered flask by passing the water in by a glass tube which reaches to the bottom, and from which it can pass by another tube just coming through the stopper which is to be air-tight. By allowing the water to run for a few minutes a fair sample is obtained. The

supply pipe is then closed by a suitable clip. In a third hole in the stopper there is a funnel with stop-cock attached for the introduction of the reagents. The first reagent is crystalline manganous chloride formed by dissolving 425.5 grammes in distilled water and diluting to 1,000 c.c. If the water sample be 500 c.c., then 2 c.c. of this reagent is introduced. The next to be added, after allowing a suitable time interval (2 to 3 minutes), is 2 c.c. of an alkaline potassium iodine solution. This is formed by dissolving

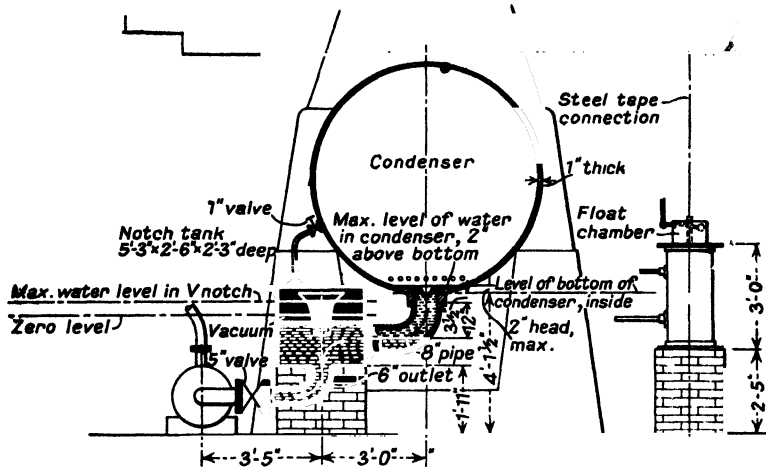
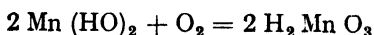
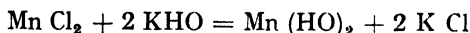


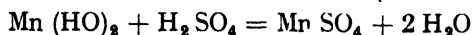
FIG. 243.

LEA RECORDER, VACUUM TYPE.

700 grammes of potassium hydrate and 150 grammes of potassium iodine in distilled water and diluting to 1,000 c.c. The result of these two additions is the precipitation of manganous hydrate which is oxidized by the contained oxygen into manganic acid, the reactions being—



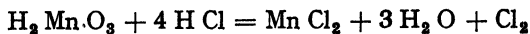
Sulphuric acid is now added, 2 c.c. of a mixture of concentrated acid and water being introduced. This dissolves any manganous hydrate not oxidized, thus—



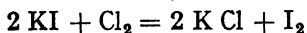
and



As a result of these actions the following interactions take place—



and



So that at this stage iodine has been liberated in a quantity proportional to the oxygen originally present in the water. If oxygen were absent, $\text{H}_2 \text{ Mn O}_3$ would not be formed, hence chlorine would

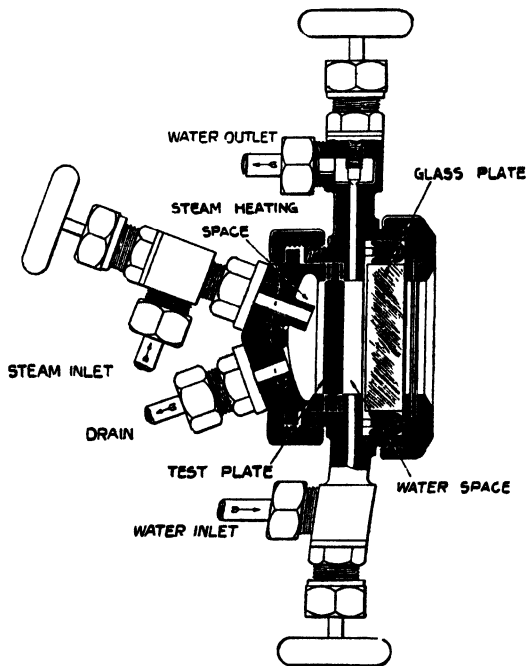
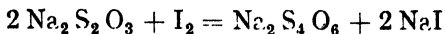


FIG. 244.

SECTIONAL VIEW OF CORROSION INDICATOR.

not be freed to attack the potassium iodine. At this stage the water can be exposed to the air as it is immune from further contamination.

To indicate the absence or presence of free iodine, a starch solution is used. This consists of 2 grammes of starch made into a paste with cold water to which is added hot water up to a volume of 100 c.c. When a few drops are added to 250 c.c. of the water any free iodine causes it to turn blue. The iodine is measured quantitatively by adding drop by drop sodium thiosulphate which causes the following action to take place—



The sodium thiosulphate is approximately .025 normal and should, preferably, be purchased from a chemist already standardized. It does not maintain its condition beyond two or three months. The volume of this reagent added is measured by admitting it from a burette, and it is added until the blue colour is discharged. If N be the volume thus added, V the volume to which the starch was added, and A the normality of the sodium thiosulphate, then the oxygen contents in cubic centimetres per litre measured at 0°C . and 760 mm is—

$$\frac{139.5 NA}{V \times 0.025}$$

Comparison with the figures on page 355 will show the relative aeration.

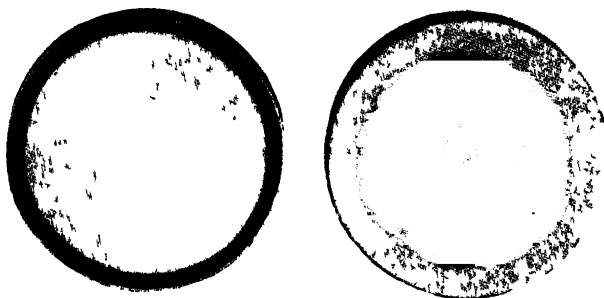


FIG 245
APPEARANCE OF DISCS OF CORROSION INDICATOR.

Portable kits containing the apparatus for making these tests have been placed upon the market by the Elliott Co., of Pittsburgh, to whom a considerable amount of this information is due, and also by Brady & Martin, of Newcastle-on-Tyne.

An apparatus has been introduced by Messrs. G. & J. Weir, Ltd., to show rapidly the condition of the water. It was noticed that the corrosive properties of water became more active if the plate in contact with the water was heated. In Fig. 244 is shown a sectional view of the indicator, from which it will be seen that the water to be tested passes between a glass plate and the test plate. The latter can be heated by steam admitted on the other side. If a closed feed system is working correctly the plate remains untarnished, but if not it becomes badly tarnished in two hours. Fig. 245 shows the appearance of the discs after this period.

STEAM

Saturated Steam. Mercury Gauges. Bar. 30" at 62° F.

Pressure		Temperature Fahrenheit		Heat in Mean British Thermal Units Fahrenheit per lb.			Volume
Vac.	Lbs.						cb.ft./lb.
ins.	abs.	<i>t</i>	<i>T</i>	<i>h</i>	<i>L</i>	<i>H_g</i>	<i>v_g</i>
29.5	0.245	58.6	518.2	26.5	1055.8	1082.3	1257
29.45	0.269	61.3	520.9	29.2	1054.4	1083.6	1150
29.4	0.294	63.8	523.4	31.7	1053.0	1084.7	1058
29.35	0.318	66.1	525.7	34.0	1051.8	1085.8	982.0
29.3	0.343	68.2	527.8	36.1	1050.7	1086.8	914.3
29.25	0.367	70.2	529.8	38.1	1049.6	1087.7	857.6
29.2	0.392	72.2	531.8	40.1	1048.5	1088.6	805.7
29.15	0.416	74.0	533.6	41.9	1047.6	1089.5	762.0
29.1	0.441	75.7	535.3	43.6	1046.8	1090.4	721.4
29.05	0.465	77.3	536.9	45.2	1046.0	1091.2	686.2
29.0	0.490	78.9	538.5	46.8	1045.1	1091.9	653.0
28.95	0.514	80.4	540.0	48.3	1044.3	1092.6	624.1
28.9	0.539	81.8	541.4	49.7	1043.5	1093.2	596.6
28.85	0.563	83.2	542.8	51.0	1042.8	1093.8	572.5
28.8	0.588	84.5	544.1	52.3	1042.1	1094.4	549.5
28.75	0.612	85.8	545.4	53.6	1041.4	1095.0	529.1
28.7	0.637	87.0	546.6	54.8	1040.7	1095.5	509.4
28.65	0.661	88.2	547.8	56.0	1040.1	1096.1	492.0
28.6	0.686	89.4	549.0	57.2	1039.5	1096.7	475.2
28.55	0.710	90.5	550.1	58.3	1038.9	1097.2	460.0
28.5	0.735	91.6	551.2	59.4	1038.3	1097.7	445.2
28.4	0.784	93.7	553.3	61.5	1037.2	1098.7	418.9
28.3	0.832	95.6	555.2	63.4	1036.2	1099.6	396.1
28.2	0.881	97.5	557.1	65.3	1035.2	1100.5	375.4
28.1	0.930	99.3	558.9	67.1	1034.2	1101.3	356.6
28.0	0.979	101.0	560.6	68.8	1033.3	1102.1	339.8
27.9	1.028	102.6	562.2	70.4	1032.4	1102.8	324.5
27.8	1.077	104.2	563.8	72.0	1031.6	1103.6	310.6
27.7	1.126	105.7	565.3	73.5	1030.8	1104.3	297.9
27.6	1.175	107.2	566.8	75.0	1030.0	1105.0	286.2
27.5	1.224	108.6	568.2	76.4	1029.2	1105.6	275.4
27.4	1.273	109.9	569.5	77.7	1028.5	1106.2	265.4
27.3	1.322	111.2	570.8	79.0	1027.8	1106.8	256.1
27.2	1.371	112.5	572.1	80.3	1027.1	1107.4	247.5
27.1	1.420	113.7	573.3	81.5	1026.4	1107.9	239.4
27.0	1.469	114.9	574.5	82.7	1025.8	1108.5	231.9
26.8	1.567	117.2	576.8	85.0	1024.5	1109.5	218.2
26.6	1.665	119.4	579.0	87.2	1023.3	1110.5	206.1
26.4	1.763	121.5	581.1	89.3	1022.2	1111.5	195.3
26.2	1.861	123.4	583.0	91.2	1021.2	1112.4	185.6
26.0	1.958	125.3	584.9	93.1	1020.1	1113.2	177.0
25.8	2.056	127.1	586.7	94.9	1019.1	1114.0	169.0
25.6	2.154	128.8	588.4	96.6	1018.2	1114.8	161.8
25.4	2.252	130.5	590.1	98.3	1017.3	1115.6	155.2
25.2	2.350	132.1	591.7	99.9	1016.4	1116.3	149.1
25.0	2.448	133.6	593.2	101.4	1015.6	1117.0	143.5

TABLES

Saturated Steam. Mercury Gauges. Bar. 30" at 62° F.

Pressure		Temperature Fahrenheit		Heat in Mean British Thermal Units Fahrenheit per lb.			Volume cb.ft./lb.
Vac.	Lbs.						
ins.	abs.	<i>t</i>	<i>T</i>	<i>h</i>	<i>L</i>	<i>H_g</i>	<i>V_g</i>
25.0	2.448	133.6	593.2	101.4	1015.6	1117.0	143.5
24.5	2.693	137.3	596.9	105.1	1013.5	1118.6	131.2
24.0	2.938	140.7	600.3	108.6	1011.6	1120.2	120.9
23.5	3.183	143.8	603.4	111.7	1009.9	1121.6	112.1
23.0	3.428	146.8	606.4	114.7	1008.2	1122.9	104.6
22.5	3.672	149.5	609.1	117.3	1006.9	1124.2	98.10
22.0	3.917	152.1	611.7	119.9	1005.4	1125.3	92.32
21.5	4.162	154.6	614.2	122.4	1004.0	1126.4	87.22
21.0	4.407	157.0	616.6	124.8	1002.6	1127.4	82.64
20.5	4.652	159.2	618.8	127.0	1001.4	1128.4	78.54
20	4.896	161.4	621.0	129.2	1000.2	1129.4	74.88
19	5.386	165.4	625.0	133.2	998.0	1131.2	68.50
18	5.876	169.2	628.8	137.0	995.8	1132.8	63.14
17	6.365	172.7	632.3	140.5	993.8	1134.3	58.60
16	6.855	175.9	635.5	143.7	992.0	1135.7	54.66
15	7.344	178.9	638.5	146.7	990.3	1137.0	51.24
14	7.834	181.9	641.5	149.7	988.5	1138.2	48.23
13	8.324	184.7	644.3	152.6	986.8	1139.4	45.56
12	8.813	187.3	646.9	155.2	985.3	1140.5	43.21
11	9.303	189.8	649.4	157.7	983.9	1141.6	41.08
10	9.793	192.2	651.8	160.1	982.5	1142.6	39.12
9	10.28	194.5	654.1	162.4	981.2	1143.6	37.40
8	10.77	196.8	656.4	164.7	979.8	1144.5	35.80
7	11.26	198.9	658.5	166.9	978.5	1145.4	34.34
6	11.75	201.0	660.6	169.0	977.2	1146.2	33.00
5	12.24	203.0	662.6	170.9	976.1	1147.0	31.76
4	12.73	204.9	664.5	172.8	975.0	1147.8	30.62
3	13.22	206.8	666.4	174.7	973.9	1148.6	29.57
2	13.71	208.6	668.2	176.5	972.8	1149.3	28.57
1	14.20	210.3	669.9	178.3	971.7	1150.0	27.65
0	14.69	212.0	671.6	180.0	970.7	1150.7	26.79
-2	15.67	215.2	674.8	183.3	968.7	1152.0	25.22
-4	16.65	218.3	677.9	186.4	966.9	1153.3	23.83
-6	17.63	221.3	680.9	189.4	965.1	1154.5	22.59
-8	18.61	224.1	683.7	192.2	963.4	1155.6	21.47
10	19.59	226.8	686.4	194.9	961.8	1156.7	20.46
12	20.57	229.5	689.1	197.6	960.1	1157.7	19.55
14	21.55	232.0	691.6	200.1	958.6	1158.7	18.71
16	22.52	234.4	694.0	202.6	957.0	1159.6	17.96
18	23.50	236.7	696.3	204.9	955.6	1160.5	17.26
20	24.48	238.9	698.5	207.2	954.1	1161.3	16.62
22	25.46	241.1	700.7	209.4	952.7	1162.1	16.02
24	26.44	243.1	702.7	211.4	951.5	1162.9	15.46
26	27.42	245.2	704.8	213.6	950.1	1163.7	14.94
28	28.40	247.2	706.8	215.6	948.8	1164.4	14.46
30	29.38	249.1	708.7	217.6	947.6	1165.2	14.01

INDEX

- ADAMSON, Daniel & Co., Muller ejector, 281
 — tests, 310
 Adiabatic and isothermal compression, comparison, 205
 Admiralty mixture, 54
 Aerated water, air contents of, 355
 Air cooling, surface for, 40
 — contents of aerated water, 355
 — in steam, effect of, 354
 —, effect of presence of, on temperature of condensation, 33
 —, heat transmission coefficient for, 38
 — pump, capacity, 175
 — —, Edwards', 220
 — -pump efficiency, variation of, 178
 — pump, hydraulic, see hydraulic air pump
 — —, Parsons compound, 223
 — —, steam ejector, see ejector, steam,
 — — reciprocating, calculations, 173
 — pumps, wet and dry, 227
 — quantities, 346 [34
 — temperature drop in condenser,
 Alberger Pump and Condenser Co., jet condensers, 151
 — Co., plant by, 349
 — Pump and Condenser tube glands, 66
 — surface condenser, 89
 Allen, S. T., 115
 Allgemeine elektricitats Gesellschaft, hydraulic air pump, 299
 Allis Chalmers Manufacturing Co., water doors, 71
 Armstrong Whitworth & Co., 134
 Atmospheric condenser, contraflo, 79
 — condensers, definition of, 2
 — —, Weir type, 83
 — pressure variations, vacuum corrections for, 3
 Austin, J., 112
 Auxiliaries, methods of driving, 304
 —, power consumption of, 9
 Auxiliary power consumption of ejector condensers, 162
 — — — — — evaporative condenser, 131
 Auxiliary power consumption of jet condensers, 139
 Axial position of jets, 327
 BALCHE (A. G.) of Bochum, 281
 Balance, heat, 309
 Barometric condensers, Alberger, 153
 — jet condenser, 140
 B.E.A.M.A. heat drop tables, 201
 — recommended air capacities, 348
 Bengough, Dr., 24
 Brass, heat transmission coefficient, 21
 Breguet ejector, 237
 —, Maison, steam ejector, 237
 Brush Electrical Co., steam ejector, 260
 Brown-Boveri continuous service condenser, 117
 — steam ejector, 268
 — feed systems, 386
 — hydraulic condensate extractor, 296
 — hydraulic ejector, 293
 — surface condensers, 97
 CALCULATION of surface area, 15
 Calculations, air pump reciprocating, 173
 —, hydraulic type, 211
 —, steam ejector, 182
 Calibration of vacuum gauge, 3
 Callendar, Prof., 22, 184
 — steam diagram, 192
 Capital expenditure, effect of varied vacua on, 6
 Centrifugal pump, effect of variable head on design, 53
 Christie, John, 111
 Clearance space, effect of, 176
 Closed-feed system, Mirrlees-Watson, 372
 — —, importance of, 358
 Coating of tubes, 112
 Cole, Marchant & Morley, steam ejector, 260
 Comines power station, 109
 Compagnie Française Thomson-Houston, 282
 Condensate, air contents of, 360
 —, cooling surface area, 41

- Condensate extraction pump, 296
 — heating, 300
 —, loss of, in steam ejector, 289
 —, loss of, in jet condenser, 138
 — meter, 388
 Conditions on starting ejectors, 324
 Continuous-service condenser, 117
 Contraflo atmospheric condenser, 79
 — closed-feed system, 364
 — hydraulic pump, 292
 — Kinetic ejector, 257
 — surface condensers, 75
 Conversion energy, 188
 Cooler, air external, 91
 Cooling water, calculation of, 13
 Corrosion electrolytic, 110
 — indicator, Weir type, 391
 — of tubes, 110
 Counter-current jet condenser, 142
 Critical water velocity, 25
 Cumberland protective system, 111
- DALMARNOCK** power station, 105
 De-aerating plant details, 361
 Definition of atmospheric condensers, 2
 — of ejector condensers, 2
 — of heat transmission coefficient, 16
 — of jet condensers, 2
 — of surface condensers, 1
 Deflection of tubes, 55
 Delas, Soc. des condensateurs, steam ejector, 260
 — — tests, 332
 — surface condensers, 106
 Determining area of diffuser, 191
 Diffuser, conditions at entry to, 186
 —, determining area of, 191
 —, mixing in, 187
 —, motion in, 187
 — throat pressure, 190
 Division of total surface, 42
 Doors, hinged inspection, 71
 —, water by Allis Chalmers, 71
 —, water load on, 72
 Dortmund Vulkan, Müller ejector, 281
 — — tube glands, 66
 — — tube plates, 60
 Dry air pump, 228
 Dryness fraction, 185
 Dual air pump, Weir, 234
- EDWARDS** air pump, 220
 Effect of air in steam, 354
 Efficiency of compression, 205
 — of steam jet, 208
- Ejector, action of increased air, 209
 — — of decreased air, 210
 Ejector capacity variation with
 — vacuum, 327
 — condensers, 162
 — condensers, definition of, 2
 — experimental designs, 241
 —, kinetic, 256
 —, Koerting Bros., 168
 —, Ljungström, 263
 —, Mirrlees-Watson, 169
 —, Rees roturbo, 170
 —, Schutte & Koerting, 163
 — steam consumption, 329
 — tests, 318
 — with water jacket diffuser, 194
 — steam, Breguet, 237
 — —, Brown-Boveri, 268
 — — calculations, 182
 — — consumption, 329
 — —, Contraflo Kinetic, 256
 — —, Delas, 260
 — — development, 237
 — — experimental designs, 241
 — —, Hick-Breguet, 237
 — —, Ingersoll-Rand, 264
 — —, Meldrum, 270
 — —, Metrovick-Leblanc, 242
 — —, Mirrlees-Leblanc, 240
 — — starting conditions, 324
 — —, Weir, 268
 — —, Wheeler C. & E. Co., 246
 — —, Wheeler Radojet, 252
 — —, Worthington, 262
 Electrolytic corrosion, 110
 Elliott de-aerating system, 378
 English Electric Co., 281
 Entraining chamber, 185
 Erosion of tubes, 112
 Escher-Wyss hydraulic pump, 280
 — surface condenser, 94
 — tube glands, 66
 — tube plates, 59
 Establishing a vacuum, 197
 Exhaust flange section, 70
 — steam for condensate heating, 307
 External air cooler, 91
 Extraction pump, Contraflo, 296
 Evaporative condenser, Ledward & Beckett, 127
 — condensers, advantages of, 130
 — —, use of, 132
- FERRULES**, 65
 Fouling of tubes, 112
 Friction loss in water boxes, 52
 —, loss of head due to, 49

Friction loss in tube ends, 52
Frictional resistance in tubes, 49
Fullerton, Hodgart & Barclay, 134

GIBSON Co, Geo H, 87
Ginabat, M A, 106

HEAD pump, 140
— — —, water head on, 144
Heat balance, 309
— — — contents of exhaust steam, 11
— — — drop tables, B E A M A, 201
— — — transmission coefficient, 38
— — — — — for air, 38
— — — — —, subdivision of, 20
— — — — — coefficient for brass, 21

Heating of condensate, 300
Hick-Breguet ejector test, 337
— — — de-aerating plant, 361
Hick-Hargreaves steam ejector, 273
— — — jet condenser, 158

Hoefer, Dr, 40
Howson, Charles & Co, 112
Hulsmeier self-cleaning condenser, 119

Hydraulic air pump, A E G, 299
— — — — —, Brown-Boveri, 293
— — — — —, calculations, 211
— — — — —, Contraflo Kinetic, 292
— — — — — Escher Wyss, 280
— — — — — Metrovick Leblanc, 271
— — — — —, Muller, 281
— — — — — tests, 370
— — — — —, Wheeler, 298
— — — condensate extractor, 296
— — — pump, Worthington, 377

INGERSOLL-Rand Co, continuous service condenser, 119
— — — ejector, 264
— — — tube specification, 93
— — — surface condensers, 91
Inspection doors, hinged, 71
Isothermal and adiabatic compression, comparison, 205

JET condenser, Alberger, 153
— — —, barometric, 140
— — —, counter current, 141
— — —, definition of, 2
— — —, Hick-Hargreaves, 158
— — —, low-level type, 140
— — —, M A N, 148
— — —, Metrovick, 147
— — —, Mirreles, 154
— — —, parallel flow, 141

Jet Condenser, Worthington-Simpson, 154
Jet condensers, auxiliary power consumption, 139
Jones, G F, 134
Josse, Professor, 40

KINGHORN valve, 220
Kinetic ejector, 256
— — — rotary pump, 292
— — — system by-pass valve, 259
Koerting Bros of Hanover, 163
— — — "Eductor" condenser 165
— — — ejector condensers, 168
Kothny, G L E, 325
Krupp, Frederick, of Kiel, 281

LANCASHIRE Electric Power Co, 338
Latent heat, 11
Lea J E, 361
Leblanc, M Maurice, 215, 241,
— — — rotary pump, 213, 271
Ledward & Beckett evaporative condenser, 127
Leeds Corporation power station, 122
Lees Professor C H, 49
Limit values of vacua, 10
Ljungstrom A B Angturbm steam ejector, 263
— — — loco condenser, 137
Loco condenser, Ramsey, 134
Loss of condensate in hydraulic air pump, 289
— — — in jet condenser, 138
Low-level jet condensers, 140

M A N jet condensers, 148
Manufacture of tubes, 54
Marine-type surface condenser, Vickers, 79
Martin, H M, 25
Martin's calculation of R3, 28
Maschinenfabrik Augsburg Nuremberg self-cleaning condenser, 119
— — — — — hydraulic air pump, 281
— — — Grevenbroich, 283
— — — surface condensers, 97
— — — tube glands, 67
Meldrum steam ejector, 270
Mercury column, temperature corrections, 342
Metropolitan Vickers Electrical Co, Ltd, steam ejector, 245
— — — surface condensers, 101
Metrovick jet condensers, 147
— — — Leblanc ejector, 242
Mirreles-Leblanc steam ejector, 240

- Mirrlees-Leblanc hydraulic air pump, 271
 ——— -Watson jet condensers, 154
 Mirrlees-Watson "central condenser" ejector, 245
 ——— closed feed system, 372
 ——— de-aerator, 377
 ——— ejector condensers, 169
 ——— jet condensers, 154
 ——— plant description, 350
 ——— surface condensers, 105
 ——— tube cleaner, 125
 ——— valve, 231
 ——— vertical condensers, 122
 Motion in diffuser, 187
 Motor or turbine drive for auxiliaries, 304
 Muller, Dr P H 281, 149
 Muller ejectors with Hulsmeyer condenser, 289
 ——— hydraulic air pump, 281
 ——— ——— tests, 310
 Musgrave John, & Sons, 252

 NON-RETURN valve, 286

 OPTIMUM value for vacuum, 6
 Orde, Cairns & Morrow, 178
 Otto Estner Dortmund 281
 Oxygen contents determination, 388

 PARALLEL flow jet condenser, 141
 Parsons' compound air-pump, 223
 Parsons, plant, by, 349
 ——— vacuum augmentor, 181, 226
 Passes, number of, 44
 Paul, J H 114
 Performance of condensers, 2
 ——— of Koerting plant, 167
 Pipework arrangement, 268
 Plates, tube, size of, 58
 Powder, C C, 74
 Pressure-control device, 295
 Pressure variation at steam inlet, 343
 Proportions of a steam jet, 198
 Protective system, Cumberland 111
 Power consumption of auxiliaries, 9

 RADOJET ejector, 252
 Ramsey Condensing Locomotive Co 134
 Ratio of diameter to length of surface condenser, 45
 Readings, recording routine, 352
 Recommended air capacity, 347
 Recording routine readings, 352
 Rees-Roturbo ejector condensers, 170
 Reynolds, Osborne, 25

 Richardsons-Westgarth, 116
 ——— surface condensers, 75
 ——— tests on plant, 340
 ——— tube glands, 65
 Riviere, P L, 360
 Rotary and ejector pumps, comparisons, 300
 ——— pump, Leblanc, 271

 SAVING due to varying vacua, 5
 Scale, effect of on heat transmission 21
 ——— ——— on condenser efficiency 115
 ——— formation, 113
 Scanes, A E Leigh, 24
 ——— pressure tank, 273
 Schuchtermann & Kremer, Dortmund, 281
 Schutte & Koerting ejector condensers, 163
 ——— ——— plant by, 351
 Shells, condenser materials, 67
 Shell, condenser thickness of, 68
 ———, stresses on, 68
 Société Anonyme de Constructions Electrique, 282
 Southend power station, 131
 Specific heat of sea water, 13
 Stabilizing device, 258
 Stanton and Pannell, 49
 Starting conditions, 305
 Steam consumption of ejectors, 329
 ——— ejector air-pumps 237
 ———, heat contents of exhaust, 11
 ——— jet efficiency, 208
 ——— ——— proportions, 198
 ———, quantity carried over by air pump, 35
 ——— tapped for condensate heating, 306
 Stodola, Professor 1 23
 Stoney, Prof G Gerald, 6
 Stresses on condenser shell 68
 Supersaturated steam, 184
 Supersaturation, 184
 Supplementary feed valve, 385
 Surface allowed for air-cooling, 40
 ——— area, calculation of, 15
 ——— ——— for cooling the condensate, 41
 ——— condenser, Alberger, 89
 ——— ——— as de-aerator, 357
 ——— condensers, Brown-Boveri, 97
 ——— ———, Contraflo, 75
 ——— ———, definition of 1
 ——— ———, Delas, 106
 ——— ———, Escher-Wyss, 94

Surface condensers, Ingersoll-Rand, 91
 — — —, Maschinenfabrik Grevenbroich, 97
 — — —, Metropolitan-Vickers, 101
 — — —, Mirrlees-Watson, 105
 — — —, Richardsons - Westgarth, 75
 — — —, Uniflux type, Weir, 81
 — — —, Vickers marine type, 79
 — — —, Wheeler, 86
 — — —, Worthington - Simpson, 83
 — — —, Worthington Pump and Machinery Corporation, 85
 — — —, division of total, 42

TAIL-pump, 140

—, water head on, 145

Temperature corrections of mercury column, 342
 — of condensation, 33

Testing for oxygen contents, 388

Test on steam ejectors, 318

— pressures for tubes, 55

Thickness of condenser shell 68

Time to establish vacuum, 316

Trap vacuum, 251

Tube-cleaner, Mirrlees-Watson, 125

— corrosion, 110

— ends, friction loss in, 52

— erosion, 112

— glands, Alberger Pump & Condenser Co., 66

— — —, electrical connection through, 66

— — —, Dortmunder Vulcan 66

— — —, Escher-Wyss, 66

— — —, Maschinenfabrik Grevenbroich, 67

— — —, Richardsons - Westgarth, 65

— — —, Worthington-Simpson, 66

— — — plates, Dortmunder-Vulcan, 60

— — —, Escher-Wyss 59

— — —, sizes of, 58

— — —, Worthington-Simpson, 59

— specifications, Ingersoll-Rand, 93

Tubes, bursting pressure of, 55

—, coating of, 112

—, deflection of, 55

—, dimensions and weights of, 43

—, fouling of, 112

—, frictional resistance in, 49

—, manufacture of, 54

—, test pressures for, 55

—, vibration of, 56

Turbine steam, variation of cost with vacuum, 8

Turbulent flow, viscous annulus in 27

UNDERTYPE condenser, Wheeler, 87

VACUA, saving due to varying, 5

—, limit values of, 10

Vacuum augments, Parsons', 181

— -breaking valve, 287

— corrections for atmospheric pressure, variations, 3

—, establishing a, 197

— gauge, calibration of, 3

—, time to establish, 316

— trap, 251

Valve by-pass, 259

—, Kinghorn, 220

—, Mirrlees-Watson, 231

—, non-return for Muller ejector, 286

—, Worthington feather, 229

Variation of air-pump efficiency, 178

Vertical condensers, Mirrlees-Watson 122

Vibration of tubes, 56

Vickers marine-type surface condenser, 79

Viscous annulus in turbulent flow, 27

WATER boxes, friction loss in, 52

— friction, loss of head due to, 49

— head on head pump, 144

— — on tail-pump, 145

—, specific heat of, 13

— passes, number of, 44

Waterworks type, Wheeler condenser, 89

Water velocity, critical, 25

Weir auxiliary condensers, 83

— closed air-feed system, 383

— condensate meter, 388

— corrosion indicator, 391

— dual air pump, 234

— steam ejector, 268

— surface condensers, Uniflux type, 81

Weiss, F. J., 71, 141

Wet and dry air pumps, 227

Wheeler Condenser & Engineering Co., 250

— condenser, Undertype, 87

— condensers, waterworks type, 89

— multiple jet ejector, 246

— plant description, 350

— radojet ejector, 252

- | | |
|--|---|
| <p>Wheeler surface condensers, 86
 —, C H Mfg Co, Radojet ejector, 252
 Willans & Robinson, 281
 Wolverhampton Corporation power station, 115
 Worthington ejector 263
 — feather valve 229
 — hydraulic pump 277
 Worthington Simpson jet condenser, 155
 — dry air pump 228
 — hydraulic air pump, 277</p> | <p>Worthington-Simpson steam ejector,
 — surface condensers, 83 [263
 — tests on plant, 337
 — tube glands, 66
 — plates, 59
 — water doors, 71
 —, wet and dry air pump, 236
 — Pump & Machinery Corporation dry air pump, 228
 — — — hydraulic air pump, 277
 — — — surface condensers, 85, 119</p> |
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